INDICATOR HANDBOOK

C. N. PICKWORTH

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INDICATOR HANDBOOK:

A PRACTICAL MANUAL FOR ENGINEERS.

BY

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PREFACE.

PUBLISHED originally in two parts, dealing respectively with "The Indicator: Its construction and Application" and "The Indicator: Its Analysis and Calculation," this work has met with a considerable measure of success.

Both parts being out of print, I have taken the opportunity of re-issuing the complete treatise in one volume, and trust it will enjoy a continuance of the favour extended to the work in its original form.

CHARLES N. PICKWORTH.

WITHINGTON, MANCHESTER.
March 1920.

THE INDICATOR HANDBOOK.

PART I.

The Indicator: Its Construction and Application.

CHAPTER I.

PRELIMINARY CONSIDERATIONS.

THE Indicator is an instrument primarily designed to give a graphic record of the manner in which the pressure of steam in an engine cylinder varies throughout the stroke of the piston. The instrument is, in fact, simply a "pressure recorder," and whether applied to the cylinder of a steam, gas, oil or hot-air engine, or to an air or water pump, its sole function is to afford an indication of the manner in which variations of pressure occur in the vessel to which it is attached. Considered more especially in connection with the steam engine, it will be seen that in order to record the pressure variation during the stroke, means must be provided (1) to measure the pressure at every instant throughout the stroke, and (2) to indicate clearly the exact position occupied by the piston at each instant of pressure measurement.

The *pressure-measuring* part of the apparatus consists essentially of a small cylinder connected to the engine cylinder, in which a light piston is arranged to move freely.

By means of a piston rod passing through the upper cover of the cylinder, and suitable mechanism, any movement of the piston is transmitted to a pencil or marking point, the arrangement generally being such that the movement of the pencil is a magnified copy of that of the piston. In the upper part of the cylinder is a spiral spring, which is so arranged as to resist any movement of the piston in either direction from its neutral position.

Upon the lower part of the cylinder being placed in communication with the engine cylinder, the pressure existing in the latter acts upon the under side of the indicator piston, while the upper side of this piston is subjected to atmospheric pressure only. It is clear that the pressure which preponderates will cause a corresponding movement of the piston to take place, the latter rising, if the steam pressure is the greater, until the increasing resistance of the spring due to its compression balances the greater pressure of the steam. On the other hand, the piston will fall if the atmospheric pressure is in excess, until the increased tension of the spring in elongating again produces equilibrium. As the compression or extension of a spiral spring is proportional to the pressure causing the change in length, it follows that—knowing the pressure required to compress or extend the spring so as to cause the pencil to move through one inch vertically—the pressure upon the piston at any instant will be indicated by the corresponding height of the pencil above the neutral position which it occupies when both sides of the piston are exposed to atmospheric pressure only.

Thus far, only the pressure-measuring portion of the instrument has been considered; it remains to indicate how the positions of the engine piston corresponding to the successive positions of the pencil are recorded. It will be evident, in the first place, that if the pencil be arranged to move in contact with a suitably-mounted sheet of paper, the varying positions of the pencil will cause a vertical line to be traced, the maximum height of which will represent the maximum pressure which has occurred in the cylinder during the stroke. But no indication of the manner in which the pressure *fluctuates* throughout the stroke is thus afforded. If, however, the mounted

sheet of paper is moved in a direction at right angles to that in which the pencil moves—and in such a manner that the motion is a reduced copy of that of the engine piston,—the pencil will no longer trace and retrace a vertical line, but will describe a curved outline representing the resultant motion due to the simultaneous movements of the pencil and paper. On the other hand, when each side of the indicator piston is subjected to atmospheric pressure only, the pencil will remain stationary in its neutral position, while the movement of the paper only, will cause a horizontal line to be traced, which, from the conditions under which it is drawn, is termed the *atmospheric line.* From this it will appear that each point in the curved figure, or *indicator diagram*, as it is called, represents by its vertical height above the atmospheric line the pressure acting upon the engine piston at that particular point in its stroke.

In nearly all modern instruments the paper upon which the diagram is traced is mounted upon a cylindrical drum or barrel, capable of turning upon a fixed stud as an axis. Motion is given to this drum by a cord passing around a groove at its base and connected to the engine in such a manner as to give to the paper a movement exactly similar to that of the engine piston, but on a much reduced scale. Since the cord gives motion to the drum in one direction only, a spring placed within the drum is employed to effect the movement during the return stroke, and this maintains sufficient tension on the driving cord to keep the drum constantly under its control.

Assuming the indicator to be attached to one end of the cylinder of an engine which is working uniformly, and that the correct movement is given to the paper-carrying drum, the figure traced by the pencil supplies a record of the behaviour of the steam while acting upon one side of the piston during a complete double stroke of the engine. Thus in the hypothetical case assumed in Fig. 1, the vertical line or ordinate 1 2 at the commencement of the stroke indicates a practically instantaneous attainment of the initial pressure, the amount of which is represented by the vertical height above the atmospheric line A L. Then the horizontal motion of the paper caused the line 2 3 to be

traced, indicating that the pressure remained uniform until the latter point was reached. At 3 the steam supply was cut off, and the consequent fall of pressure, combined with the continued movement of the paper, produced the curved line 3 4. Release occurred at 4, the vertical line 4 5 representing an instantaneous fall of pressure and marking the termination of the forward stroke. The motion of the paper now being reversed, the line 5 1 was drawn, representing by its height above the atmospheric line A L, the amount of pressure (supposed constant in this case) which opposed the movement of the engine piston during its return stroke. At 1 the cycle of operations is again commenced by the fresh admission of steam to the cylinder.

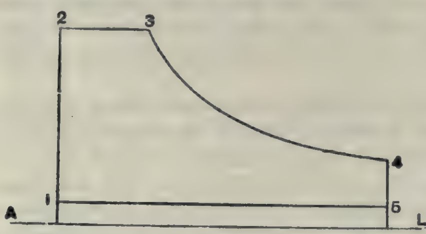

FIG. 1.

It will be shown, subsequently, that from the area of the indicator diagram, the power developed by the engine may be ascertained, this application of the instrument constituting one of its principal uses. Another and perhaps more important function of the indicator is to show the distribution of steam in the cylinder, by indicating the manner in which the admission, cut-off, expansion, release, and compression of the steam takes place, thereby acting as an invaluable guide to the correct adjustment of the valve gear. The indicator diagram also shows whether the ports and passages for the admission and exhaust of the steam are of sufficient size and suitably arranged. Cylinder condensation and re-evaporation, leakage of steam past the piston or valves, and (in conjunction with feed water measurements) the weight of the steam passing through the engine in a given time, are among other important

items of information which may be deduced by the intelligent study of the indicator diagram.

Nor are the useful applications of the indicator restricted to the engine cylinder. Applied to the steam chest, or to the steam and exhaust pipes, the instrument is frequently found of great value in determining the resistance offered by the pipes and ports to the passage of the steam. Its application to the receivers of compound engines should be mentioned; while in condensing engines, the application of the indicator to the condenser allows a direct comparison to be made between the amount of back pressure shown by the diagram and the pressure in the condenser. Again, by attaching the indicator to the air pump, the power required to operate this apparatus may be determined; while much useful information may be obtained by applying the instrument to air compressors, blowing engines, refrigerating plant, etc.

In connection with pumping plant, the indicator furnishes information in regard to the resistance offered by pump valves, the utility of air vessels in reducing pressure fluctuation in pipe lines, etc. Applied to machinery operated by hydraulic or pneumatic power, many useful data are obtained as to the power required for various operations. In fact there is scarcely any limitation to its useful application wherever fluid pressure variations are to be measured and recorded.

CHAPTER II.

THE CONSTRUCTION OF THE INDICATOR—STEAM ENGINE INDICATORS.

IN common with many of the most important adjuncts of the steam engine, the indicator was invented by James Watt. Few particulars of the earliest form of apparatus are available, doubtless owing in a great measure to the fact that Watt kept his valuable invention a secret for many years. There is, however, reason for believing that in its original form the instrument consisted merely of a cylinder about 6in. in length and 1in. in diameter, which was fitted with a piston. A rod attached to the latter, and passing through the upper end of the cylinder, was provided with a pointer; while between the upper face of the piston and the cylinder cover was placed a long spiral spring, the ends of which were attached to the piston and cover respectively. The upper side of the piston was open to the atmosphere, while the lower part of the cylinder communicated with the engine cylinder. With the low speeds usual at this period, some idea of the action of the steam could be obtained by closely watching the movement of the pointer; but this crude device was useless as a power-measuring and recording apparatus, and it was not until Watt added the sliding paper-carrying board—representative of the motion of the piston—that the principle of the present form of indicator was established. This sliding board was arranged to move in a direction at right angles to that of the piston, and was actuated by a cord attached to a suitable point in some reciprocating part of the engine, the return movement being effected by a weight attached to another cord acting in the opposite direction to the driving cord.

Early Instruments.—Watt's indicator rendered very

great service in the early days of steam engineering, when low speeds and low steam pressures were employed. But with increase of engine speed the necessity for modifying the apparatus soon became apparent. It was found that the reciprocating board which carried the paper, and the cords and counterweights used to actuate it, rendered the instrument totally unsuitable for indicating any but very slow-moving engines.

With a view to remove this defect, McNaught introduced the indicator bearing his name, and the improved instrument enjoyed considerable favour for many years. In this indicator a hollow vertical cylinder, capable of rotating about its axis, was substituted for the sliding board previously employed, and the paper, wrapped around this cylinder, was secured thereto by means of a clip. The driving cord passed round a groove at the foot of the paper barrel, and was secured to the latter. A guide pulley, by means of which the driving cord could be led into the groove in the barrel, was also added by McNaught.

It will be readily understood that upon the cord being pulled, the paper-carrying drum will rotate, and a diagram will be traced, as previously described, provided means are employed to ensure the prompt return movement of the drum. This was conveniently effected by means of a flat coiled spring contained within the base of the drum, which, by keeping the cord constantly in tension, formed a very efficient substitute for the balance weight employed by Watt. The extent of the motion of the drum was limited by means of stops to about three-fourths or seven-eighths of a complete revolution.

The arrangement of steam cylinder, spring, etc., was substantially the same as that adopted by Watt, except that the pencil was carried by a short arm secured to the piston rod near the lower end of the spring, and projecting through a slot in the upper part of the cylinder.

Slightly modified forms of the McNaught instrument were subsequently made by Maudslay and others in this country, and by Stillman in America, but the only one which came at all into general use was the indicator made by Messrs. Hopkinson, of Huddersfield. In the original form of this instrument, the paper-carrying drum surrounded the lower

part of the steam cylinder, the axes of the two coinciding. As in the McNaught indicator, a short arm was fixed to the piston rod at the lower end of the spring, and from this depended a long pencil rod, the lower end of which carried the pencil. In a later form of this instrument, the more usual drum arrangement was reverted to, the pencil being carried by a horizontal arm mounted upon the top of the piston rod, and capable of swivelling thereon in order to adjust the position of the pencil with regard to the paper.

In order to satisfactorily fulfil its purpose, the indicator must produce a diagram sufficiently large to enable the action of the steam to be shown distinctly, and experience soon led to the recognition of the fact that a diagram of from 2 to 3in. in height was necessary to ensure successful results. It follows that in all direct-acting indicators—wherein the piston and pencil are rigidly connected—a spiral spring of considerable length must be employed, which, upon the admission of high-pressure steam to the indicator cylinder, will be thrown into vibration, the extent of the resulting oscillations of the pencil being greatly intensified by the momentum of the reciprocating parts. The effect produced upon the diagram by this vibratory action depends greatly upon the speed of the engine. With slow-running engines the oscillations of the pencil may almost entirely cease before the paper has moved perceptibly; but with higher speeds the oscillations distort the diagram very considerably, the pencil tracing a series of undulating curves, which, being in no way due to the action of the steam within the cylinder, seriously detracts from the value of the indicator diagram.

There are other disadvantages attending this arrangement of indicator. The compression of the spring is frequently accompanied by a bending action which has the effect of tilting the piston and producing an undue amount of friction. Further, the extensive motion of the piston operates against a simultaneous indication being obtained of the variations of pressure which are occurring within the cylinder.

What appears to be the earliest attempt to obviate these disadvantages was made by Sir Daniel Gooch about 1840. In the form of instrument which he devised specially for

use on locomotives, the piston moved in a horizontal cylinder rigidly fixed to a framing, while the springs were curved flat strips, similar in form to a carriage spring. An unequal-armed lever was used, the end of the longer arm carrying a pencil, while the end of the short arm was connected to the piston rod by means of a short link. The paper used in this instrument was in the form of a long strip, being continuously unwound from one roller on to another, and at a rate proportional to the speed of the engine The pencil moved through a considerable arc, and from the motion of the paper it will be clear that the diagram traced will not form a closed curve, but that indications of successive strokes will be given in one continuous line. The atmospheric line was traced by a second pencil fixed to the frame, and by means of a curved scale the more usual form of diagram could be readily constructed. This indicator, which is said to have worked well up to 400 strokes per minute, is worthy of notice as being the first instrument in which a lever was employed for the purpose of amplifying the motion of the pencil.

Richard's Indicator.—It is not a little surprising that the lever movement used by Gooch was not generally adopted. No doubt the curvilinear movement of the pencil constituted the principal objection to the arrangement, but it is remarkable that more than twenty years elapsed before this defect was remedied. The introduction in 1862 of the instrument designed by Mr. C. B. Richards, of Hartford, Conn., U.S.A., marks a distinct epoch in the history of the indicator. This instrument, representing the earliest form of the modern indicator, was in every respect a very great improvement upon any hitherto constructed, and at the present time it is still extensively used for moderate speeds.

Of the accompanying illustrations, Fig. 2 shows the instrument partly in section, while Fig. 3 represents the indicator as made by Messrs. Elliott Bros. From Fig. 2 it will be seen that a short spring D is employed, the upper end of which is attached to the cylinder cover E, while the lower end is secured to the piston B, the latter being arranged to move with perfect freedom in the liner or bushing fixed in the steam cylinder A A. The upper end of the piston rod is coupled by a short link to a point G in the upper bar

H I of the parallel motion. An exactly similar bar M L, and a centre link I L, completes the parallel motion, the arrangement being such that the pencil K describes an approximately straight line, the extent of the movement being four times that of the piston, since the length H G is one-fourth of H I. For the usual travel of the pencil, the

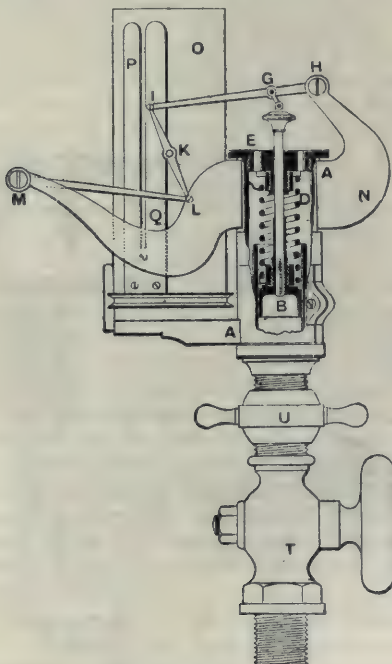

FIG. 2.

stroke of the piston is thus diminished considerably, and a much shorter and stiffer spring may be employed, resulting in a corresponding reduction of elastic vibration. The two fixed centres of the parallel motion are carried by suitable brackets projecting from, and forming part of, a cylindrical sleeve which fits upon the outside of the steam cylinder, and around which it can turn freely. The joint at the

top of the piston rod is arranged to swivel, and this allows the sleeve, together with the parallel motion and pencil, to be rotated upon the cylinder, enabling the pencil to be brought into contact with or withdrawn from the paper as desired.

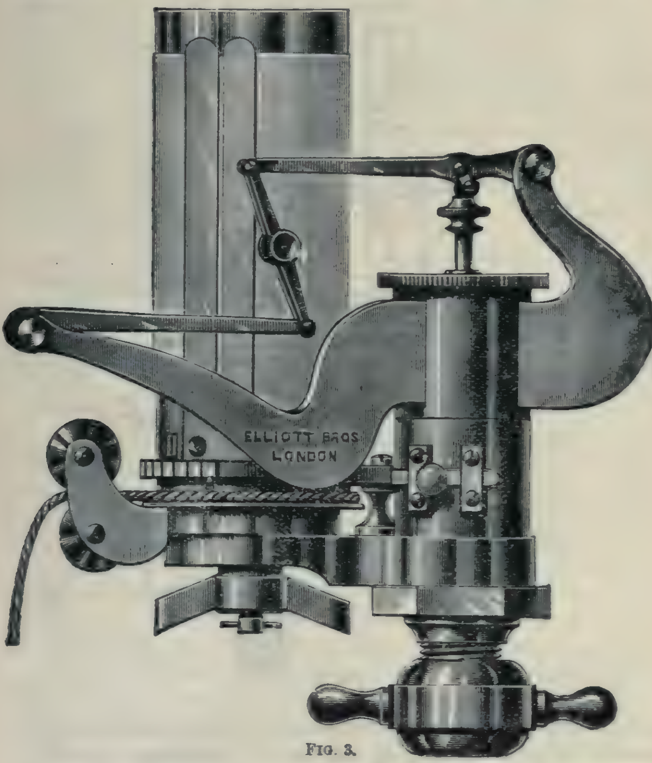

FIG. 3.

The diameter of the piston in this indicator is 0·7978in., giving an area of 0·5sq. in The extreme travel of the pencil is 3in. The piston, of a specially hard metal, is an easy fit in the cylinder, in order to ensure a movement free from friction; while, to prevent an accumulation of

steam above the piston, large escape holes are provided in the cylinder cap, as shown in Fig. 2.

At the lower part of the instrument a coupling U is provided in order to connect the indicator with the stop-cock T. The lower extremity of the stem of the instrument is conical in form, and fits into a correspondingly coned seat in the stop-cock, and as the thread upon the indicator and that upon the stop-cock are of different pitches, the rotation of the coupling draws the cone firmly into its seat, forming a very effectual and conveniently-made joint. The lower end of the stop-cock is screwed with a $\frac{3}{4}$-in. Whitworth thread.

The diameter of the paper-carrying drum O is 2in., giving a maximum length of diagram of 5in.; a length of $4\frac{1}{2}$in. will, however, generally be found sufficient. In the place of the usual form of pencil, a pointed brass wire is employed to draw the diagram upon specially prepared metallic paper. A split clip P Q is provided, by means of which the paper is readily attached to the paper drum. The grooved pulley upon which the cord is coiled, and the guide pulleys, are clearly shown in Fig. 3.

The "detent" device shown applied to the paper drum in Fig. 3 offers a convenient means for instantly stopping or starting the motion of the paper drum without disconnecting the driving cord. It consists of a small pawl mounted on a short standard, and so arranged that it may be thrown in or out of gear with the teeth of a ratchet segment on the base of the paper drum, this being conveniently effected by the short spring partly encircling the cylinder and provided with a small knob as shown.

In order to adapt the indicator to the various pressures now used, springs are made of strengths ranging from 10 to 250lb. per square inch. When it is required to change the spring, the small coupling is unscrewed from the end of the piston rod, and the cover from the cylinder. The spring may then be unscrewed from the piston and the cover, the new spring inserted, and the parts replaced.

The Thompson Indicator.—In the Richards instrument a greatly improved action was obtained by diminishing the motion of the piston, and consequently reducing the length and vibration of the spring. But with a still further increase

in engine speeds, the momentum of the parallel motion exercised a disturbing influence upon the diagram, and a further reduction of the weight of the mechanism was found to be necessary.

To meet this requirement, Mr. J. W. Thompson, of Salem, Ohio, U.S.A., introduced in 1875 the improved instrument which bears his name, and of which Figs. 4 and 5 are illustrations of the form made by the American Steam Gauge Company, the original makers of this instrument.

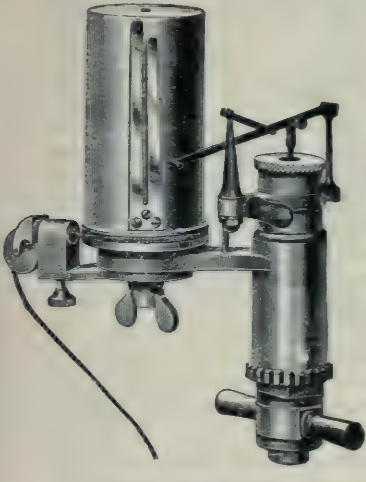

FIG. 4.

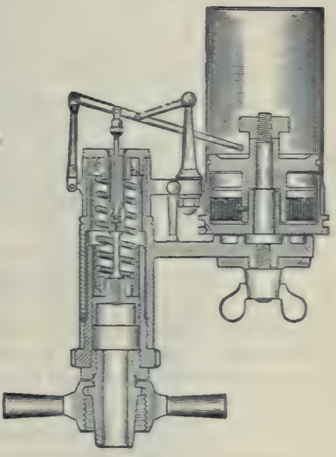

FIG. 5.

It will be observed that the parallel motion differs essentially from that used in the Richards instrument, the single lever being reverted to and the pencil caused to move in a straight line by a modified arrangement of the Scott-Russell parallel motion.

From the illustrations it will be seen that the fulcrum of the pencil arm, being placed at the upper end of a swinging link, is free to move laterally. A short and light radius bar has one end attached to a point in the pencil arm and the other to a fixed standard, as shown, the length and point of attachment being so arranged that the curved

path which the pencil would have described if the fulcrum had been fixed, is counteracted by the effect of the short radius rod, and the pencil describes an approximately straight line. The link connecting the piston rod and pencil arm is fitted at the lower end with a modified form of ball-and-socket joint, which provides for the lateral movement required by the parallel motion, as well as for the rotative movement which occurs when the pencil arm is moved to or from the paper drum. Means are provided for taking up any lost motion in the joints resulting from wear or other causes.

This construction of parallel motion reduces the weight of the moving parts to a very considerable extent, the downward pull at the pencil due to this weight in the Thompson indicator being less than one-third of that in the Richards instrument; but the reduction of disturbing effect is still greater, since the pencil arm, not having to carry other links, may be made very much lighter, and this results in a very material reduction of the weight of those parts which move at a high velocity, rendering the instrument available for speeds up to 400 revolutions per minute. The links, pencil bar, etc., have been reduced in weight to the lowest limit consistent with the requirements of every-day use; while a considerable improvement has been effected by placing the drum spring close to the base of the drum, thereby lessening the weight and reducing the spindle friction.

During recent years indicators with external springs have come largely into favour, and nearly all the leading makers now supply instruments of this type. As will be shown later, the strength of a spring is influenced by its temperature. Hence, when a spring calibrated in air under normal conditions is used in an indicator cylinder, it is necessary to make an allowance for the effect of the increase in temperature to which it is subjected. The more satisfactory method adopted by many makers is to calibrate the spring at the temperature at which it is to be used. It will be obvious, however, that while this temperature may reasonably be taken at about 212° Fah. for the ordinary conditions of steam practice, it may be considerably higher when the instrument is used with superheated steam or on a gas or

oil engine. But by so locating the spring that it is unaffected by variations of temperature, any uncertainty of this kind is avoided altogether. The spring can be more accurately calibrated under normal conditions, and more readily tested either by dead weight or steam pressure if

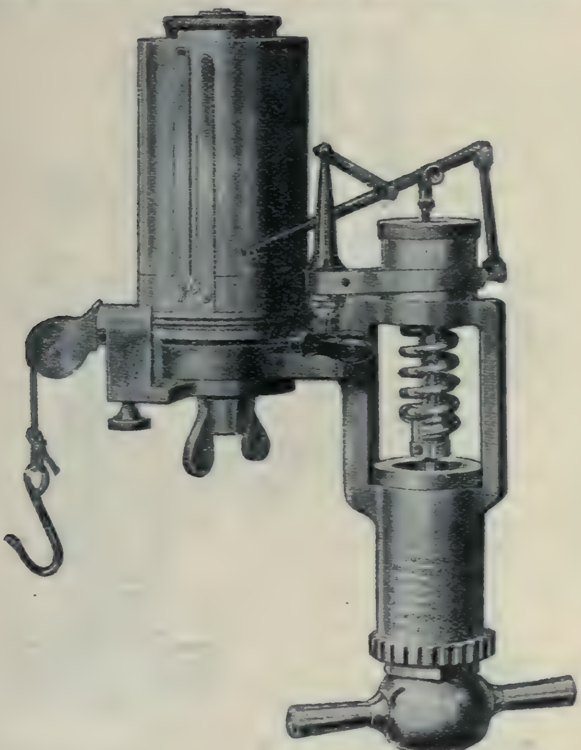

FIG. 6.

necessary at any time. Minor, but not insignificant, advantages are found in the greater comfort with which the springs can be handled, and in most instruments, the greater facility with which they can be changed.

These conditions are approximately fulfilled in the instru-

ment shown in Fig. 6, which represents a very recent form of Thompson indicator made by the American Steam Gauge Company. As will be seen, the spring is exposed and is kept cool by the air circulating around it. In other respects the instrument is of standard design.

In the Thompson indicator, as made by Messrs. Schäffer

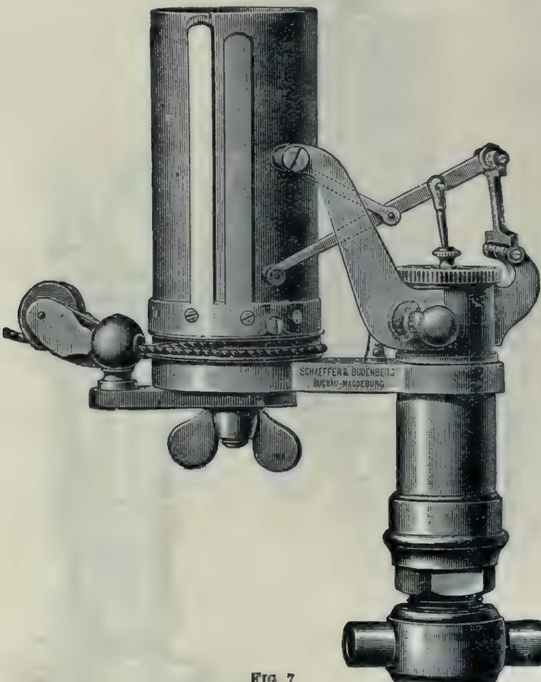

FIG. 7.

and Budenberg, Ltd., of Whitworth Street, Manchester, and Magdeberg-Buckau, the parallel motion is modified so as to avoid the hollow piston rod used in the original form, a shorter connecting rod being employed. The indicator shown in Fig. 7 is suitable for pressures up to 250lb. per square inch and for speeds up to 400 revolutions per minute, while diagrams 3in. high and 5in. long may be obtained if

desired. For speeds up to 600 revolutions the makers supply a smaller pattern, giving diagrams $1\frac{3}{4}$in. high and $3\frac{1}{2}$in. long.

Fig. 8 shows a form of Thompson indicator by the same

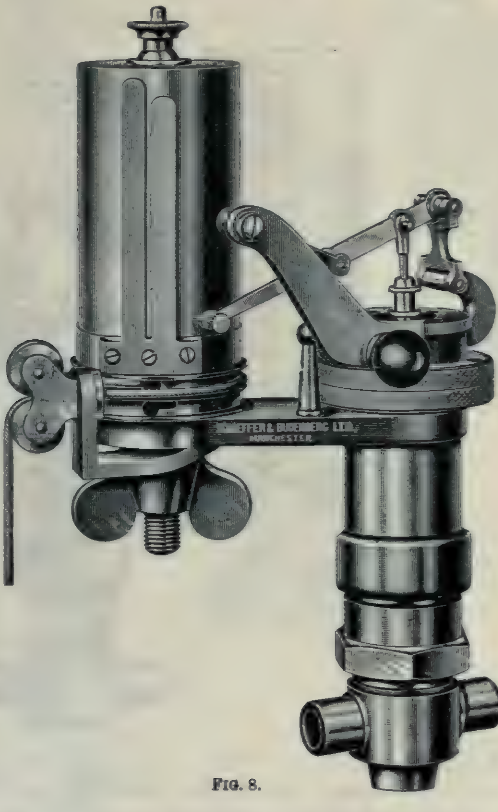

FIG. 8.

makers and known as the "1911" pattern. The piston, of 0·5sq. in. area, is of steel, and the instrument is available for pressures up to 350lb. per sq. in. The cylinder liner is surrounded by steam and can be readily removed for

cleaning. The springs are double-wound and made with a

FIG. 9

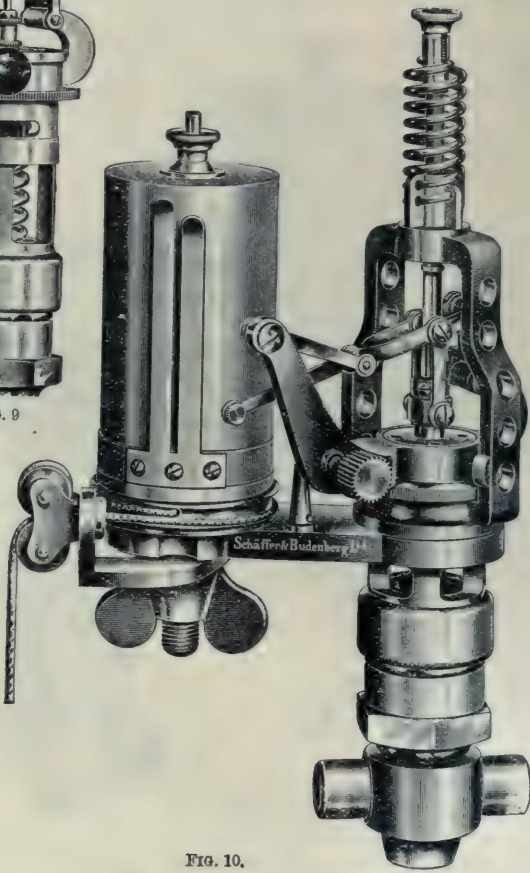

FIG. 10.

ball which forms a ball joint between the piston and piston rod. The movement of the piston is multiplied six times

at the pencil. The drum is ordinarily made $1\frac{5}{8}$in. diameter by $2\frac{1}{4}$in. high, for speeds up to 600 revolutions. By using an alternative cylinder and piston of $\frac{1}{4}$sq. in. area, the instrument can be used to 700lb. per sq. in.

In another form of the same instrument, the cylinder casing is cut away (Fig. 9) so that the spring is exposed. With this arrangement the spring is cooled by the circulating air, while accumulation of pressure above the piston is impossible.

In the Thompson indicator with inside spring, shown in Fig. 10, the spring is attached to a bridge connected with the cylinder cover, and the spring is extended instead of being compressed. The springs are tested and adjusted in a cool state, and hence difficulties due to different temperatures of the springs are entirely avoided. Accumulation of steam pressure above the piston due to leakage past the piston is prevented by the large openings provided in the cylinder casing above the piston. The indicator is made with a piston of $\frac{1}{2}$sq. in. area and is available in four sizes, for 400, 500, 700, and 1000 revolutions per minute respectively.

A very convenient instrument is the "Willner" outside spring indicator, shown in Fig. 11. In this instrument the spring is screwed on to a boss on the cylinder cover, and the Thompson pencil movement is inverted, the pencil arm being forked at the suspended end so as to clear the spring. To remove the spring it is only necessary to unscrew the milled nut at the top of the piston rod, when, by tilting it over to the left, the spring can be unscrewed from the boss on the cylinder cover. If the cover is now unscrewed, the piston and piston rod can be withdrawn. The cylinder cover is insulated by a vulcanised fibre disc, and it is found that the spring remains quite cool even during prolonged tests. The pencil mechanism is very strong and rigid; but, owing to the very compact design, the weight of the moving parts is exceptionally small for an outside spring indicator.

All these instruments are provided with a stop motion for regulating the pressure of the pencil upon the paper. Threaded into a projection from the rotating sleeve which carries the pencil motion is a screw, the point of which

comes into contact with a small fixed post or stop when the pencil approaches the paper. By regulating the position of this screw, a very minute adjustment of the pencil pressure may be readily effected. The screw is provided with an insulated handle, which is also of service in rotating the

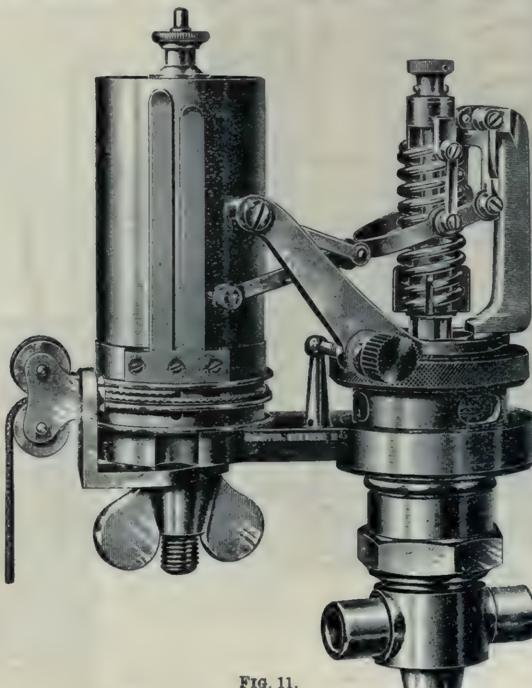

FIG. 11.

pencil mechanism to and from the paper when using the instrument.

The Tabor Indicator.—With a view to still further reducing the weight of the pencil mechanism, Mr. Harris Tabor, of New York, introduced in 1879 another form of indicator, in which the straight-line movement of the pencil is obtained in a manner which constitutes a radical departure from the systems previously employed. Fig. 12 is

a sectional illustration of the instrument on a scale of two-

FIG. 13.

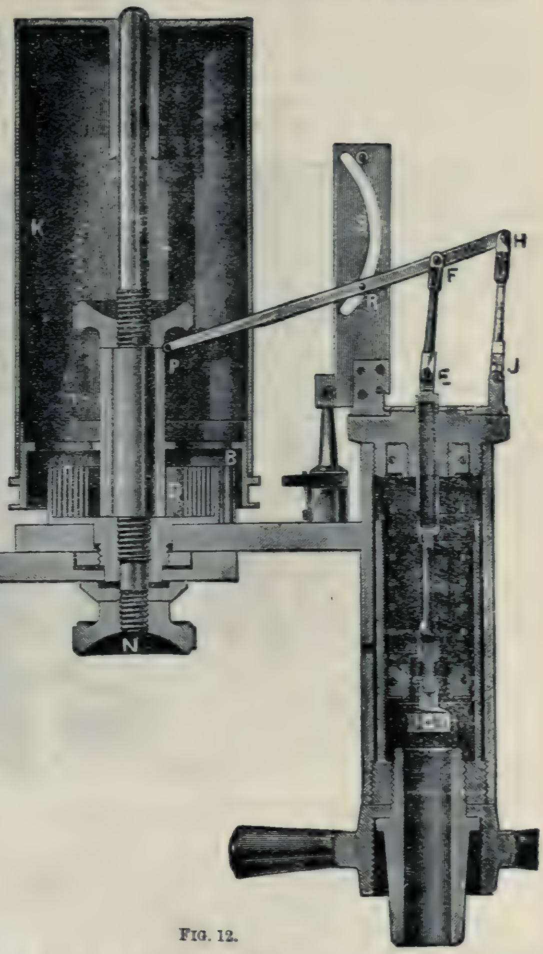

FIG. 12.

thirds full size. P H is the pencil lever, J H its swinging fulcrum, and E F the connecting rod. At a suitable point

in the pencil lever a pin is fixed, upon which a small roller R is arranged to turn freely, while in a vertical plate secured to the cylinder cover a curved slot S is formed, in which the roller R is arranged to travel, the curvature of the slot and the position of the roller being arranged so as to cause the pencil P to describe a straight line. The outer curve of the slot is approximately that of a circle having a radius of 1in.

From the construction it will be seen that the links H J and F E always remain parallel to each other; that a straight line may be drawn through the points J, E and P in every position; and also that if an imaginary link, parallel to F H, be supposed to connect the point E with the link H J, the arrangement would form an exact pantagraph.

The upper part of the steel piston rod is hollow, having an external diameter of $\frac{3}{16}$in. The lower part is solid, reduced in diameter, and provided at the end with a ball, forming part of a ball-and-socket joint, by means of which the piston and rod are connected. The socket is an independent piece, which fits into a square hole in the piston, and is secured by means of a central threaded stem and thumb-nut C. Shallow grooves, cut upon the outside of the piston, tend to keep it steam-tight by the so-called "water packing." The steam cylinder proper is separate from the outer casing, the space between the two preventing excessive heating of the exterior of the instrument.

The duplex type of spring used in this indicator (Fig. 13) consists of two spiral coils of wire, secured at the ends to suitable winged nuts. The springs are mounted so that the points of attachment of the two coils are upon opposite sides of the nut. By this means the side strain induced by the bending of one coil of the spring is exactly counteracted by that of the other, and the piston is kept in a central position in the cylinder. To enable the pencil to be brought into contact with the paper drum, the cylinder cover is fitted with a swivel plate upon which the pencil mechanism is mounted, the attachment being made by means of the small standard J on the one side, and by the slotted vertical plate on the other. From Fig. 14 it will be seen that the slotted plate referred to is backed by another plate of the same size, but without a slot. This serves to receive the

pressure to which the pencil bar is subjected during the taking of the diagram, and also keeps the pencil bar in place. The pressure of the pencil on the paper is regulated by the screw shown passing through the lower part of the slot plate, the end of the screw striking against a stop. The screw is provided with an insulated handle, sufficiently

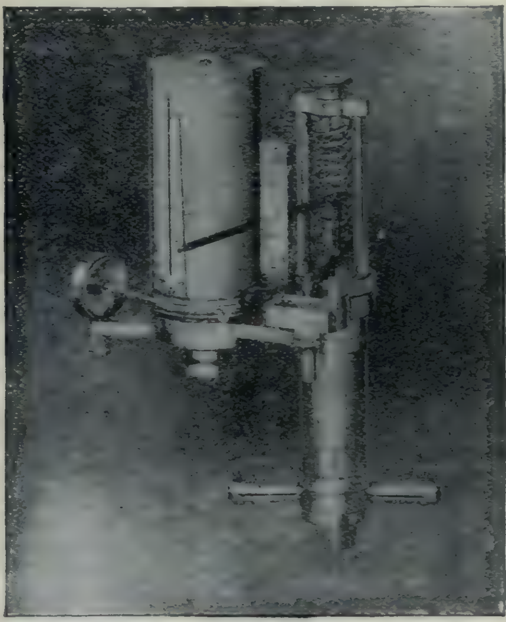

FIG. 14.

long to be readily operated by the fingers, and also serving as a handle for turning the pencil mechanism as required.

The paper drum K is made of very thin brass tube, closed at the top in order to afford suitable strength; the carriage upon which the drum is mounted has a long bearing upon the vertical shaft on which the drum rotates. The drum

spring Q, which is of the watch-spring type, has one end secured to the circular box in which it is contained, while the other is fixed to the long sleeve of the carriage. The stop B, which limits the travel of the paper drum, is shown to the right of the spring box. In order to adjust the tension of the drum spring, the thumb-nut N is loosened, when the carriage may be raised clear of the stop, and the tension adjusted by winding or unwinding as desired.

The form of guide pulley used is simple and ingenious. The pulley G is contained within a vertical disc or sheave, this disc being held by a circular frame arranged to swivel upon a pivot, which may be locked in any position by means of a nut as shown. By this device the cord may be readily guided in any direction required.

The piston of the Tabor indicator is of the usual diameter, 0·7978in., corresponding to an area of 0·5sq. in. The stroke of the paper drum is 5·5in., its diameter 2·063in., and its height 4in. The extreme range of motion of the pencil is 3·25in., and as the pencil mechanism multiplies the piston motion fivefold, the extreme travel of the piston is 0·65in. For very high speeds an instrument is supplied with a paper drum 1·5in. in diameter, 2·8in. high, and with a stroke of 4in.

Fig. 14 shows a Tabor indicator with the spring outside. The thrust of the spring is taken by a yoke attached to columns fixed in the cylinder cover as shown.

The Darke Indicator.—The method adopted by Mr. E. T. Darke for producing the rectilinear movement of the pencil is shown very clearly in Fig. 15, which gives a general view of the instrument as made by Messrs. Elliott Bros. In this case a single light steel pencil lever is pivoted at one end upon fixed steel centres as shown. For some distance from the fulcrum the lever pencil is circular in section, and is fitted with a small steel sleeve, which can slide easily upon the lever, but without lost motion. The sliding sleeve swivels in the fork at the upper end of the piston rod, and is therefore enabled to accommodate itself to the varying angular positions of the pencil lever.

In order to render the lever somewhat elastic the greater part of it is reduced in thickness, the requisite strength being provided by increasing the depth. Upon the outer

end of the lever is a sliding carriage, which is constrained to move in a straight line by sliding in a slot formed in a vertical guideplate, and carries the pencil in the form of a metal pin. As in the Richards instrument, the piston

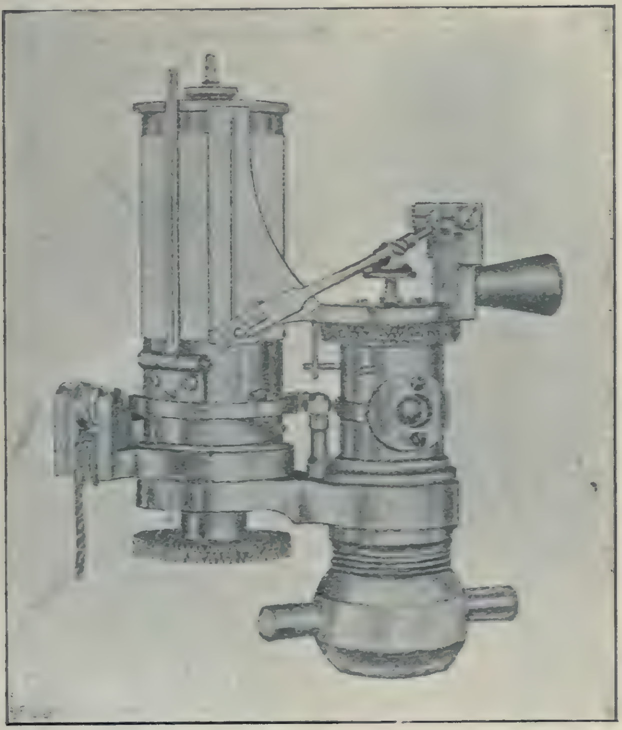

FIG. 15.

movement is multiplied four times by the pencil mechanism. The marking point is kept in contact with the paper by the elasticity of the lever, and both the guideplate and the standard carrying the fulcrum centres are mounted upon a socket piece, which may be rotated by means of the

handle shown, in order to bring the pencil in contact with the paper. This indicator will give a diagram 3in. long and $1\frac{3}{4}$in. high; it is specially suitable for high-speed work.

The Crosby Indicator.—This instrument which was invented by Mr. G. W. Brown, of Boston, U.S.A., is manufactured by the Crosby Steam Gage Company of that city, and 147 Queen Victoria Street, London. A sectional view is given in Fig. 16, and an enlarged view of the spring in Fig. 17. Referring to the sectional view, it will be seen that the pencil lever 16 is jointed at 18 to the swinging fulcrum rod 13, a link 14 connecting the piston rod and pencil lever. At a point 20 in this connecting link is coupled a short link 15 turning on a fixed centre at 21. By this linkage the point 19 receives sufficient lateral movement to cause the pencil 23 to move in a straight line. In this instrument the point 19 is so placed that the range of movement of the pencil is six times that of the piston. The whole of the pencil mechanism is carried by the sleeve 3, of which the arm X also forms a part. This sleeve can be turned on the upper part of the cylinder A by the insulated handle 22. This latter is threaded through the arm, and its point abuts against a stop fixed in the baseplate, for the purpose of adjusting the pressure of the pencil on the paper in the manner previously described. The piston 8, which is of the usual diameter—0·7978in., giving an area of 0·5sq. in.—has shallow grooves on its outer surface, which assist in maintaining steam-tightness, while diminishing friction.

The spring used in the Crosby indicator (Fig. 17) is made of a single piece of wire wound into a double coil, the upper ends of the two coils being screwed into the four wings of the nut as shown. By this method of construction the strength of the spring may be exactly adjusted by screwing the wire in or out of the nut as required. The foot of the spring consists simply of a small steel ball brazed on to the centre of the straight portion of the wire uniting the two coils of the spring. This ball replaces the heavy brass head to which the lower end of the ordinary spring is soldered, thus effecting a very great reduction of the weight of that portion of the spring which virtually forms a part of

the piston, and in which a diminution of weight is of the greatest importance.

The spring is connected to the piston through a ball-and-socket joint, which allows the spring freedom of motion, while the connection of the piston and rod is effected in the following manner:—A boss on the piston is screwed

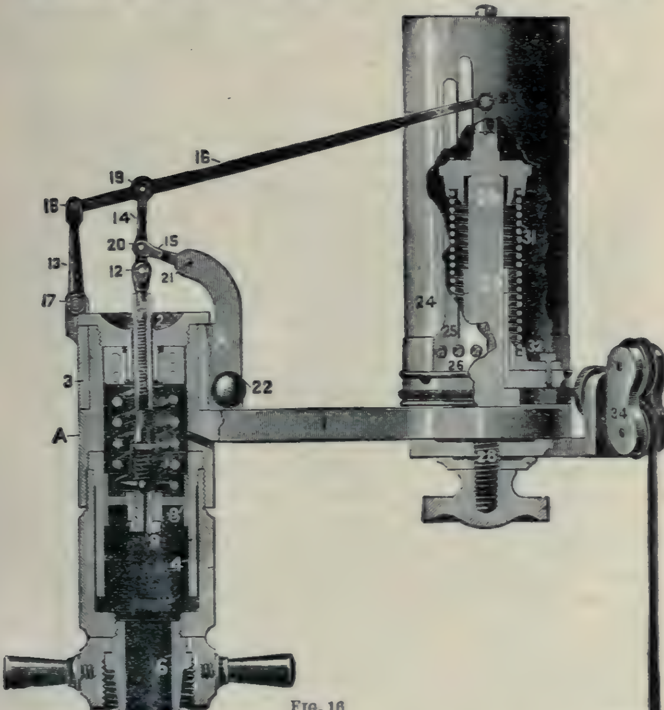

FIG. 16.

internally to receive the lower end of the hollow piston rod 10, while diametrically across this boss a slot is cut, extending down to the body of the piston, and of a width sufficient to allow the straight part of the spring to enter freely. Another boss on the underside of the piston is fitted with a set-screw 9, the upper face of which is formed with a

V-shaped recess, providing the lower seating for the ball on the spring. The upper seating is formed in the end of the piston rod, which is first screwed into the slotted boss, and all slackness taken up by adjusting the set-screw 9.

The upper part of the tubular piston rod receives the rod 11, which is threaded for the greater part of its length. By screwing this rod in or out of the tube 10, the total length of the piston rod, and consequently the position of the atmospheric line on the paper, may be altered as found convenient.

Another distinguishing feature of the Crosby indicator is the special form of drum spring employed. In place of the volute spring usually adopted, a short spiral spring 31 is used, the arrangement being clearly shown in Fig. 16. The lower end of the spring is attached to the base of the paper drum, and the upper end to a collar which fits on the squared top of the spindle 28. This collar may be raised above the squared part of the spindle, and the tension of the spring readily adjusted to suit the various engine speeds by turning to the right to increase or to the left to decrease the tension. In the standard instrument the paper drum 24 is 1½in. in diameter, but for low-speed work the makers fit a 2-in. drum if desired.

FIG. 17.

Crosby External-Spring Indicator.—This instrument, known as the Crosby "New" indicator, and shown in Figs. 18 and 19, presents several novel features. The spring, which is mounted on a strong cross bracket, has the usual Crosby ball connection with a long hollow rod, the latter being slotted to admit the pencil mechanism, as shown clearly in Fig. 19. The piston, attached to the lower end of this rod, is 1sq. in. in area, and takes the form of the central zone of a sphere. This gives only a line contact with the cylinder, and as the rod passes freely through the cylinder cover and upper guide, the piston is able to accommodate itself to any slight eccentricity in the movement of the spring. But this action is without effect upon the pencil mechanism, as the latter is independently connected to the piston by the inner rod shown in Fig. 18. This rod, which has a ball-and-

socket connection with the piston, slides through a sleeve attached to the base carrying the pencil mechanism, and thus ensures the movement transmitted being perfectly axial. The ease with which the spring can be removed and replaced is a good feature.

The makers are introducing an indicator similar to the instrument just described, but with a piston of 0·5sq. in.

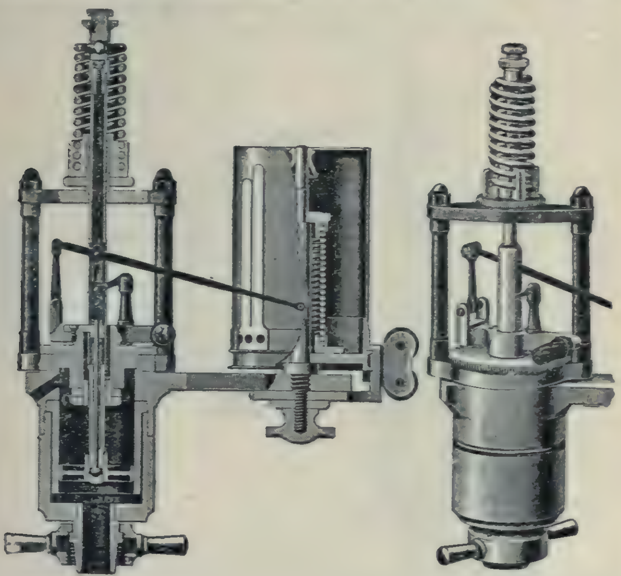

FIG. 18. FIG. 19.

area. This is known as the Crosby "New" Indicator, No. 2.

Crosby Indicator with Continuous Drum.—Fig. 20 shows a Crosby "New" indicator fitted with a continuous drum for taking a series of diagrams, and of particular service for indicating winding engines, rolling mill engines, etc. The outer shell of the drum has an opening cut in it about $1\frac{1}{4}$in. wide, and centred within this space is a pin upon which a roll of paper is mounted. Within the shell, and

concentric with it, is a smaller cylinder connected to the main drum. The roll of paper (2in. wide and 7ft. or 12ft. in length) is placed on the pin within the opening, passed round the outer shell of the drum, and in through the opening to the inner cylinder to which it is attached. The paper is advanced by pawls and a ratcheted collar on the top of the drum, an adjustment being provided by means of which it is possible to vary the number of diagrams, taken on each foot of paper, from 6 to 100. When the

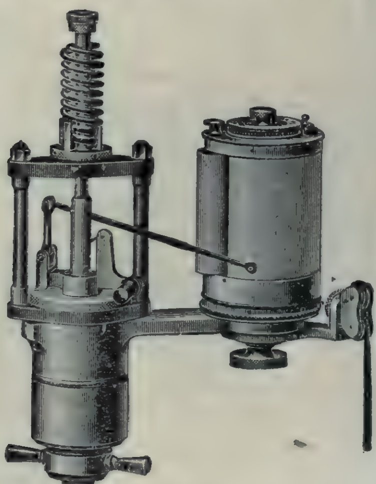

FIG. 20.

complete roll of paper has been wound on the inner cylinder, the latter may be removed through the top of the drum after disengaging and removing the knurled knob shown. A pencil mounted in a slotted bracket near the opening in the drum continues the atmospheric line throughout the length of the paper, the position of the line having been first determined by pulling the drum round by hand, and the stationary pencil adjusted to continue it. A detent is provided on the top of the drum, allowing the indicator to be used as an ordinary instrument.

The Casartelli Indicator. — This strongly made and

convenient instrument, made by Messrs. J. Casartelli & Son, Market Street, Manchester, is shown in Fig. 21. It has a pencil motion of the Crosby type, made of hardened steel and giving a sixfold multiplication of the piston motion. The piston and rod are of hardened steel and ground; a

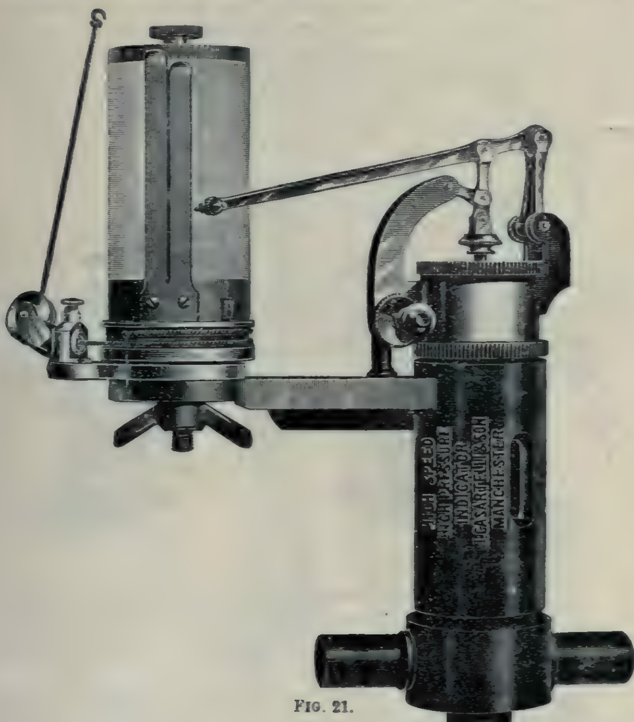

FIG. 21.

ball joint at the base of the rod prevents binding of the piston in the cylinder and ensures a free movement. The cylinder liner is isolated from the body of the instrument, and the latter has an opening formed on one side, as shown in the illustration. This prevents overheating and enables the piston to be lubricated while in use. For convenience

in handling the body of the indicator, the underside of the stage and the coupling are sheathed in ebonite. The paper drum has a spiral spring, with convenient means for adjusting the tension on the cord. The instrument shown is for high-speed engines, and springs for pressures up to 240lb. per sq. in. are supplied. The larger indicator, of the same pattern, is suitable for speeds up to 150 revolutions per minute and for pressures up to 300lb. per sq. in.

The same makers supply a modified form of the Richards indicator, fitted with a steel piston having a ball joint connection with the rod. An opening is formed in the

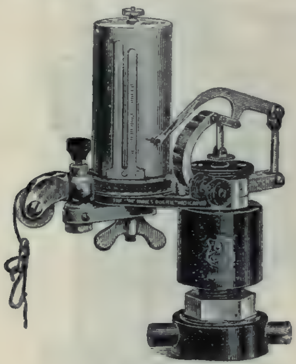

FIG. 22.

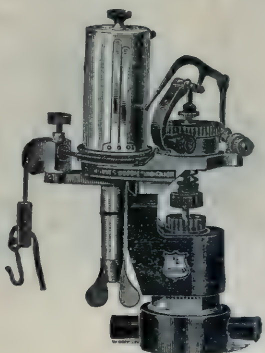

FIG. 23.

cylinder side, as in the Casartelli instrument, and the pencil motion is much lighter than in the old form, strength being secured by making the parallel motion of steel, hardened and tempered.

Dobbie-McInnes Indicators.—Of these instruments, made by Messrs. Dobbie, McInnes, Limited, Bothwell Street, Glasgow, several designs are available. The standard enclosed-spring type is shown in Fig. 22, and the standard external-spring type in Fig. 23. A sectional view of the latter instrument given in Fig. 24 shows the construction very clearly. It will be understood that the pencil arm C, which is cranked to clear the supporting bracket M, has its two centres and the marking point in one straight line,

which latter is to be regarded as the virtual pencil arm of the mechanism. The two levers give a combined multiplying movement to the pencil equal to six times that of the

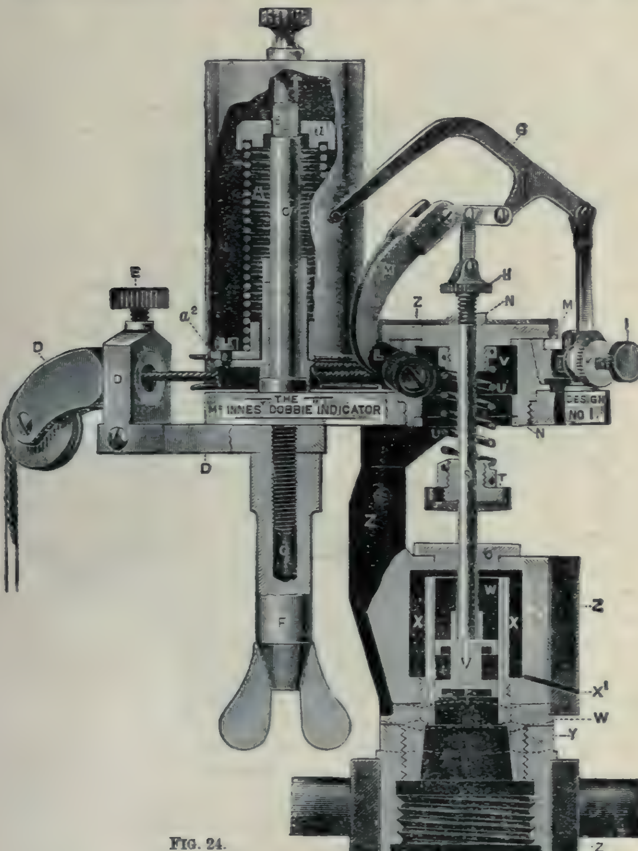

FIG. 24.

piston without the use of a long pencil arm. The piston V is of steel, and is formed of two hollow discs with a space between which will contain lubricant, and also provide a space

for grit, so avoiding scoring of the cylinder liner W. Any steam passing the piston finds its way through the annular space X, and is discharged in a downward direction, through the escape hole X′. To remove the piston, the upper cover N is unscrewed and also the cylinder cover O. The swinging bracket M, spring U, piston rod S, and piston V can then be withdrawn. The pressure of the pencil on the paper is regulated by the screw I, while the bracket M is turned by means of the vulcanite-covered arm L. The parts marked Z are sheathed in vulcanite, adding greatly to comfort in handling the instrument.

The spiral drum spring A has brass ends a^1 and a^2. The latter is fixed to the drum, while the former engages with a square B on the drum spindle C. To alter the tension of the spring, the milled head is unscrewed from the top of the spindle and the drum removed. The end a^1 is then drawn off the square, turned to the right or left, and replaced on the square B.

This instrument is made in three sizes. The largest, suitable for speeds up to 250 revolutions, gives a card $4\frac{1}{2}$in. long and $2\frac{1}{2}$in. high; while the small size, for speeds up to 800 revolutions, gives a card 3in. long and $1\frac{3}{4}$in. high. What is known as the "half size," is supplied for speeds up to 1,500 revolutions per minute, giving a card 2in. long and 1in. high.

The Dobbie-McInnes Continuous Indicator.—A special form of external-spring indicator for taking diagrams continuously from engines having a varying load is shown in Fig. 25. The paper drum contains a cylinder A on which is a roll of paper 9ft. long. One end of this is brought to the outside between the rollers 1 and 2, carried round the drum, re-entering between rollers 3 and 4 to the spindle B. On the top of the latter is a pinion gearing with a wheel on the under side of the rack C, and arranged to turn in one direction only. When the cord pulls the drum outwards, the two pinions turn with it, but remain stationary on the return stroke, and hence wind the paper from A to B as shown by arrows. The travel of the paper is about $\frac{1}{2}$in., so that the second diagram is placed $\frac{1}{2}$in. in advance of the previous one, each diagram being complete in itself. The process continues automatically until the roll of paper is

exhausted. By raising the rack lever C the instrument can be used for taking ordinary single diagrams.

The "Cipollina" Dobbie-McInnes Indicator.—In this in-

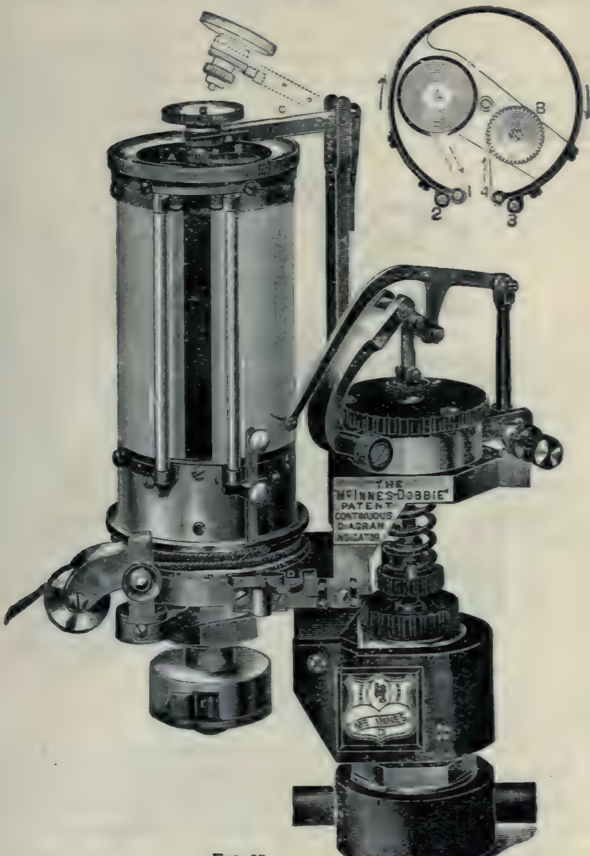

FIG. 25.

strument (Fig. 26) two indicator cylinders are employed, both acting on the same paper drum, the latter being of the continuous form already described. The two indicator

cylinders are connected with the two ends of the engine cylinder, and the drum actuated in the usual manner. The movement of the latter causes the spindle E to rise, and by means of the cam F and ratchet wheel G, the cam wheel H is slowly rotated. When projecting points on the rim of H reach the pawl I, the latter, by means of the connecting link K, carries the two pencil points L to the paper, and as the paper drum is reciprocating and the pencil arms rising and falling at each stroke, one pencil gives a diagram from

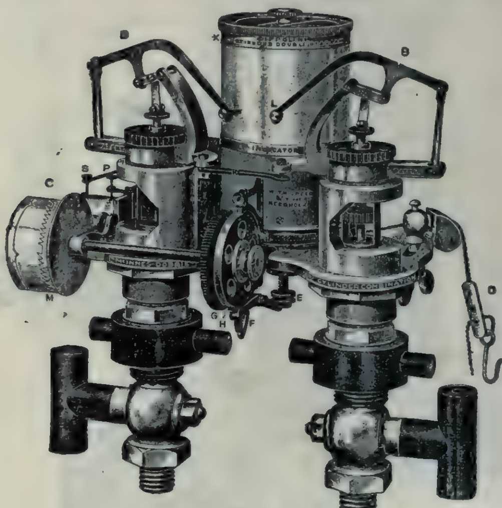

FIG. 26.

the top end of the cylinder and the other a diagram from the bottom end, both being taken simultaneously and on the same length of paper. Immediately the diagrams are taken the pencil points are released, and the same mechanism now causes the length of paper occupied by the diagrams just taken to be drawn forward into the drum, leaving a fresh portion of the same paper ready for the next set of diagrams. This goes on automatically until the paper is exhausted. The cam H, which determines the intervals at

which the diagrams are taken, can be replaced by others,

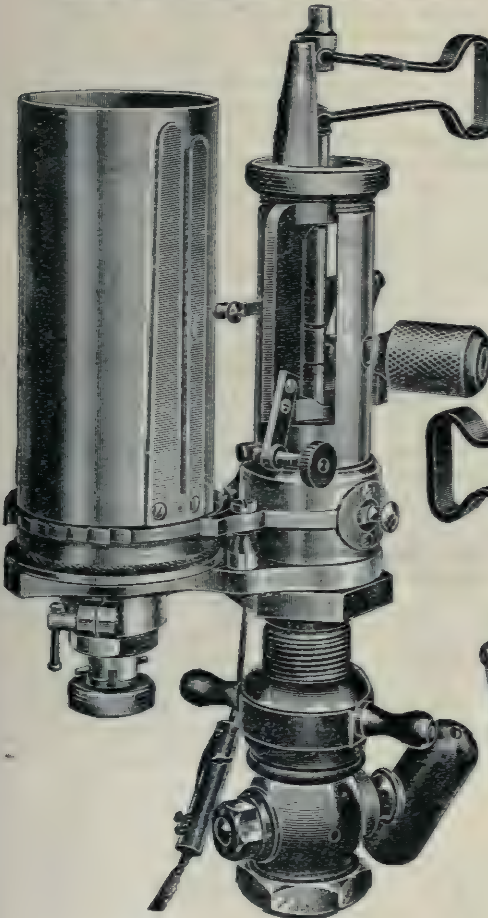

FIG. 27.

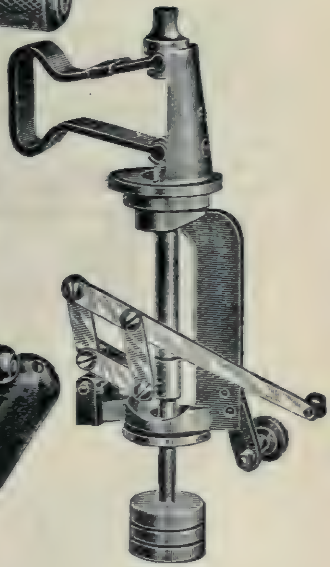

FIG. 28.

so that cards can be taken, automatically, at intervals of 25, 50, or 100 revolutions.

A revolution and time recorder is also included in this instrument. The spindle of the wheel G carries a paper drum M over which a strip of paper is led after passing under the pencil P. This latter receives a slight movement at each stroke of the large paper cylinder, producing the serrated record as shown. As the pointer is arranged to miss at every tenth stroke, an interval appears on the record which enables the result to be read in multiples of 10. To register the time, an electrical connection with a clock is arranged to actuate the second pencil S, the record, which may be in intervals of 30 seconds, being shown on the other edge of the paper strip as seen on the left of M.

The Simplex Indicator.—This outside-spring indicator exhibits a departure from usual practice in the form of the spring, which is "tongs-shaped." As shown in Fig. 27, the method of application is both simple and convenient. The spring can be readily inserted and is quite out of the influence of heat and steam. The makers (Messrs. Elliott Bros.) also claim that this form of spring lends itself to accuracy of calibration to any range.

By removing the milled nut at the top of the instrument, the piston, piston rod, and pencil motion can be entirely removed, as shown separately in Fig. 28. From this view it will be seen that a simple pantagraph pencil motion is employed. A minor, but much appreciated, feature is the provision of the insulated handle shown, by which the instrument can be held in comfort while hot. This instrument is made in two sizes, of which the larger gives diagrams up to 3in. high and 5in. long; while the small size gives diagrams up to 1½in. high and 3in. long.

CHAPTER III.

INDICATORS FOR GAS AND OIL ENGINES, ETC.

ALTHOUGH well adapted to meet all ordinary requirements of modern steam practice, it is found that the standard instruments, when used on gas and oil engines, do not by any means give equally satisfactory results, the sudden shock which the indicator piston receives at the instant of explosion, and which is transmitted to the levers of the pencil movement, not infrequently resulting in these being strained and distorted to a very serious extent. This has led to the introduction of special instruments for indicating internal-combustion motors, in which the links and levers of the pencil motion are of more substantial construction, and with these satisfactory results are obtained.

With an ordinary indicator the high pressures met with in gas engine work would necessitate the use of much stronger springs than are usual in steam engine practice, and to obviate this gas engine indicators are usually furnished with a piston of one-half the standard area, that is, 0·25sq. in. in place of 0·5sq. in. For very high pressures, pistons of 0·125sq. in. in area, or even smaller, are available.

It has now become customary for makers to supply combination indicators, the cylinder having two bores to receive pistons of 0·5 and 0·25sq. in. respectively. In this way, while the instrument is available for steam engine work, it can be readily converted into a gas engine indicator, by attaching the smaller piston. The same springs are used in both cases, but it will be understood that, when the small "gas engine" cylinder and piston are used, the strength of the spring is, in effect, doubled, and, consequently, if the pressure is read on the diagram by the scale corresponding to the normal strength of the spring, this reading must also be doubled.

The Casartelli Gas Engine Indicator.—This indicator, shown in Fig. 29, has a strong pencil motion giving a movement to the pencil of four times that of the piston.

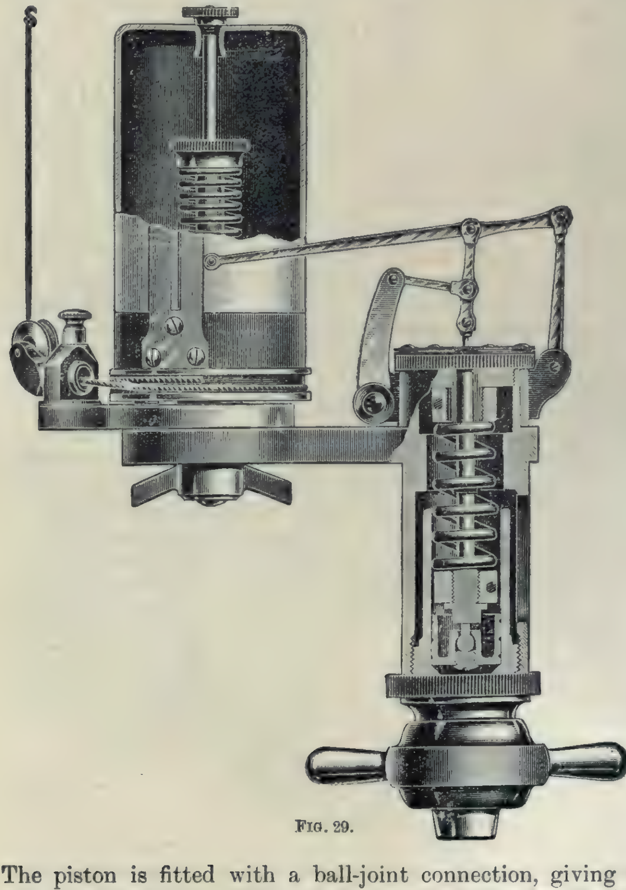

FIG. 29.

The piston is fitted with a ball-joint connection, giving a free moving piston. The cylinder cover is fitted with flat spring cushion plates, to which a steel sleeve is attached, passing down inside the cover. When the piston reaches its

upper limit of movement a shoulder on the piston rod strikes the lower end of the sleeve, so that the shock due to

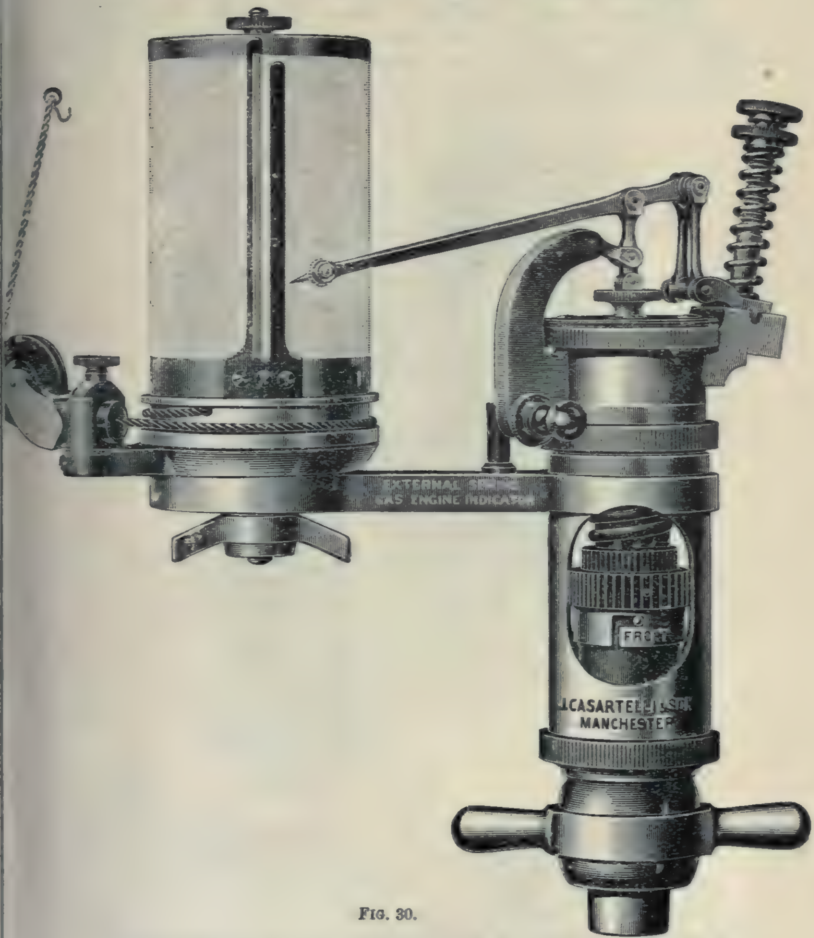

FIG. 30.

any excessive pressure is transmitted to the cushion plates, which absorb it and prevent injury to the instrument.

The piston usually furnished is 0·5sq. in. in area, but half-size pistons can be supplied and also combination sets.

Atkinson's Spring Relief Gear. — Fig. 30 shows an indicator fitted with Atkinson's Patent Spring Relief Attachment, designed to relieve the instrument of the excessive momentary shocks met with in indicating gas

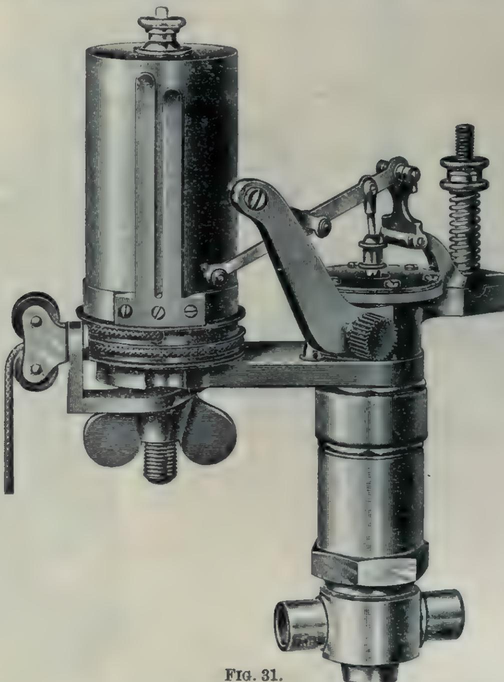

FIG. 31.

engines. The swinging fulcrum rod is not connected directly to the indicator but is carried in a stirrup held in contact with a bracket by a strong spiral spring. The tension of the spring can be regulated by the milled nuts shown. In addition to relieving the pencil movement of shock, the attachment lessens the vibratory oscillations, which in gas engine diagrams are sometimes considerable.

The Thompson Gas Engine Indicator.—This instrument, made by Messrs. Schäffer & Budenberg, Limited, is shown in Fig. 31. It is fitted with Atkinson's spring relief gear and also with the spring cushion plates, as previously described. The piston has an area of 0·25sq. in., but an

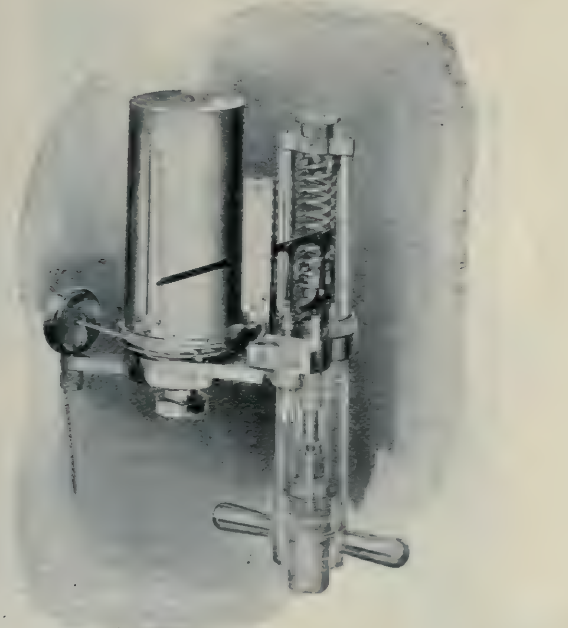

FIG 32.

additional piston of 0·5sq. in. area can be fitted to enable the instrument to be used for ordinary steam engine work.

The Tabor Gas Engine Indicator.—As shown in Fig. 32, this instrument is a combined steam and gas engine indicator, the small piston (0·25sq. in.) operating in the upper part of the cock tube.

The Crosby Gas Engine Indicator.—The standard gas

engine indicator made by the Crosby Steam Gage and Valve Company is similar in external appearance to the steam engine indicator (Fig. 16) except that the pencil lever is of a stronger section, combining lightness with stiffness, and a stronger connecting link is provided. The

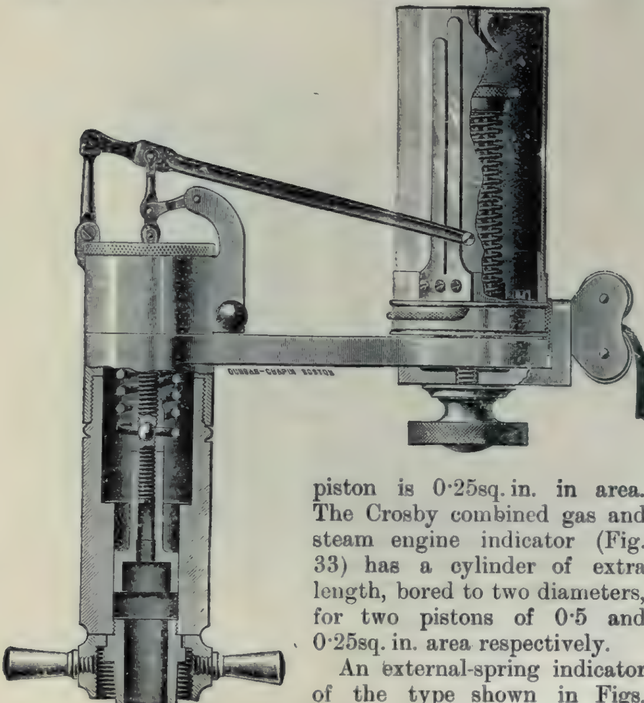

FIG. 33.

piston is 0·25sq. in. in area. The Crosby combined gas and steam engine indicator (Fig. 33) has a cylinder of extra length, bored to two diameters, for two pistons of 0·5 and 0·25sq. in. area respectively.

An external-spring indicator of the type shown in Figs. 18 and 19 is also supplied for gas engine work; the piston in this case is 0·5sq. in. in area, but the springs are rated proportionally so that the scale of the spring gives the scale of the diagram, as in steam engine practice.

Dobbie-McInnes Gas Engine Indicators.—For indicating gas and other explosive engines, Messrs Dobbie, McInnes, Limited, supply an external-spring indicator similar

to that shown in Fig. 23, but with a piston 0·3989in. in diameter, giving an area of 0·125sq. in. It is suitable for speeds up to 800 revolutions per minute, and gives diagrams $3\frac{1}{4}$in. long by $1\frac{1}{2}$in. high. The instrument is also made with a magazine drum accommodating a continuous roll of paper. For very high speeds a "half-size" instrument is made. This gives diagrams up to 2in. long and 1in. high, and is suitable for speeds up to 2000 revolutions per minute.

In the same makers' combined gas and steam engine indicator, a spare cylinder is supplied, which can be screwed into the indicator cylinder. In this instrument the steam cylinder is 0·25sq. in. in area, and the supplementary piston, 0·125sq. in. With the steam cylinder in use the springs are read at their rated strengths, these being doubled when the gas engine piston is in use.

The Mathot Explosion and Pressure Recorder. — This apparatus, shown in Fig. 34, attached to a Dobbie-McInnes gas engine indicator, gives a graphic record showing the number of explosions, the initial pressure of each, the order of their succession, together with the irregularity or otherwise of the variations, the number of revolutions and the proportion of "misses," the rate of compression, etc. This record is in point of fact a continuous succession of indicator diagrams, so greatly foreshortened that the distance from one upward movement of the pencil to the next is only about $\frac{1}{16}$in. The apparatus is of service in determining the best proportion of air and gas or oil, while the effects of varying the compression, speed, point of ignition, dimensions of inlet and exhaust valves, etc., are also made evident.

In Fig. 34 the recording drum A, $3\frac{3}{4}$in. in diameter, carries a band of paper and is rotated by clockwork within the drum, B being the winding key. The rotation of the drum can be stopped at any time by pressing the knob D. In this recorder the drum makes a complete revolution in two minutes and is restricted to speeds under 300 revolutions per minute.

In the instrument shown in Fig. 35 the band of paper, 8ft. 6in. long, is contained in a cylinder A, from which it is led over the drum B, and re-wound on the cylinder C. Gear wheels at D can be adjusted to regulate the rate of feed of

the paper, which can be varied from 8in. to about 50ft. per minute. The winding key is at N. As will be seen, the pencil arm passes between the cylinder of the recorder and the paper drum of the ordinary indicator, enabling continuous and ordinary diagrams to be taken as required.

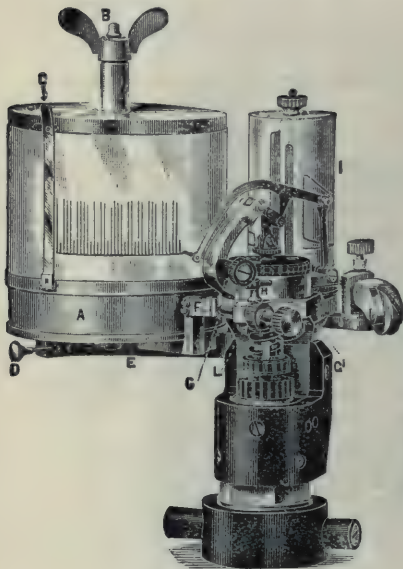

FIG. 34.

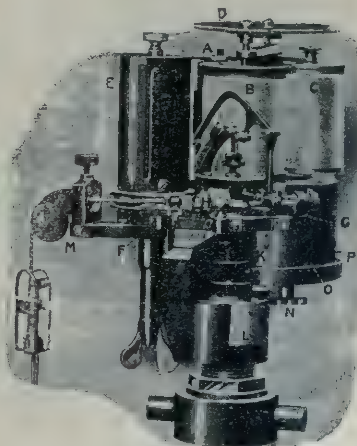

FIG 35.

SPECIAL FORMS OF INDICATORS

In addition to the standard types of indicators already described, there are a few special instruments which call for consideration.

The Hopkinson Flashlight Indicator.—In the various optical indicators, which have been introduced from time to time, a thin diaphragm of steel usually takes the place of the piston and spring of the standard instrument. The slight movements of the diaphragm caused by the variations

of pressure are communicated to a pivoted mirror upon which a beam of light is allowed to fall. The reflected beam thus records the pressure fluctuations, and, as the mirror also receives motion in a direction at right angles to the first and proportional to the movement of the engine piston, the line of light traces out an indicator diagram.

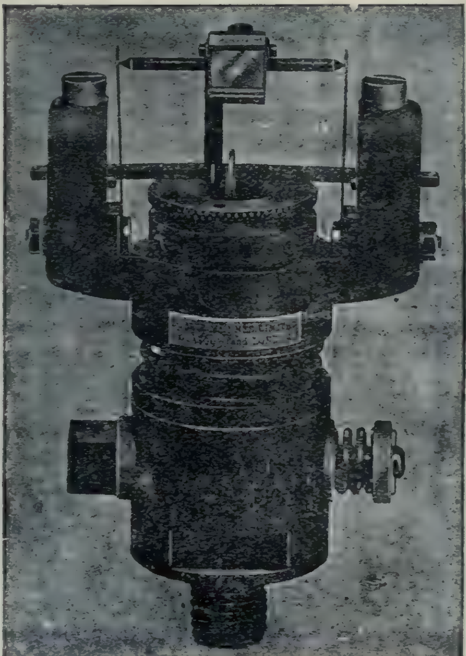

FIG. 36.

In the "Hopkinson" indicator, shown in Fig. 36, a piston and spring are used, the latter taking the form of a steel strip passing over the top of the cylinder as shown in the illustration, the ends being fixed to the upper frame of the indicator by screws. The piston moves in the bore of the indicator body and is provided at the top with a hook

embracing the spring, but in such a manner as to allow the piston to move laterally in order to prevent binding action. Three pistons are used, the areas being in the ratios of 1, 2, and 4; and, as there are two springs, the strengths of which are in the ratio of 1 to 5, a wide range of sensitiveness is obtained. The smaller pistons fit inside liners inserted in the main bore.

The mirror is clamped to a steel spindle, the ends of which are pivoted in small holes in the light spring supports shown. The movement of the spring is communicated to the mirror by a flat vertical spring attached to an arm on the mirror spindle, this spring, while sufficiently rigid to transmit the motion without buckling, being flexible enough to allow for the slight angular motion of the arm.

The upper frame of the indicator is bored to fit over the body of the instrument; it is held up to the cylinder cover by the coil spring shown, a ball race being interposed so as to allow the frame to turn easily around the axis of the instrument. A light lever clamped to the frame receives motion from a reducing gear, and in this way the frame is given a slight angular oscillation around the body of the instrument, this corresponding to the drum movement of the ordinary indicator.

The resulting movement of the line of light (which in high-speed engines forms a continuous figure) can be inspected through a telescope arrangement provided with a graduated screen, or photographed for future reference. As will be seen, the instrument has the advantage that the spring can be calibrated accurately, and when in use it is not affected by temperature variations; further, the effects of the inertia of the piston, pencil movement and paper drum are enormously reduced. It is therefore particularly adapted for indicating internal combustion engines, high-speed steam engines, etc. Messrs. Dobbie, McInnes, Limited, are the makers.

Other Optical Indicators.—Among other optical indicators of recent introduction, mention may be made of the Hospitallier-Carpentier Manograph and the "Clerk" optical indicator. In the latter a piston and spring of the ordinary type are employed. A chain, replacing the piston rod, gives a slight rocking motion to the mirror spindle, the

latter being under spring control to keep the chain taut. The line of light is received on a sheet of sensitised paper held in a frame sliding in a direction at right angles to the piston movement and actuated by a reducing gear.

Other Types of Engine Indicators.—Another method by which inventors have sought to overcome the distortion due to the momentum of the moving parts of piston indicators is by restricting the vertical movement within small limits, and gradually changing the position of these limits. In this way the indicator card is made up of a series of diagrams of small vertical height. The Wayne indicator, introduced by Messrs. Elliott Bros. (who have now discontinued its manufacture), gave diagrams of this kind. In Ripper's Mean-pressure Indicator, made by Messrs. Schäffer and Budenberg, two pressure gauges record the mean forward and back pressures acting in the cylinder to which the instrument is attached, the difference of these readings (subject to a small percentage correction) giving a very close approximation to the mean effective pressure as ordinarily determined from the area of the indicator diagram.

Indicators for Ammonia Compression Machines.—Owing to the corrosive action of ammonia upon brass and bronze alloys, the instruments used in connection with ammonia refrigerating machinery are invariably made of iron or steel. For this service, steel indicators of the ordinary type are furnished by the Crosby Steam Gage and Valve Company and other makers. As leakage past the piston is objectionable when dealing with gaseous ammonia, a pistonless indicator, such as that made by Dreyer, Rosenkranz and Droop, of Hanover, possesses advantages for this work. In this instrument a corrugated steel disc about 2in. in diameter is used in place of the usual piston, and as the movement of the centre of this plate is only slight (about $\frac{3}{32}$in.), the connecting link, which communicates this movement to the pencil lever, is attached to the latter at such a point that a multiplying movement of about twenty times is given to the pencil.

Indicators for Hydraulic Work.—Another form of pistonless indicator introduced several years ago is that known as Kenyon's Indicator. Although only indifferently successful

when used for indicating steam engines, it has found a useful present-day application in connection with such high pressures as are met with in hydraulic work. In this instrument, of which several forms are made by Messrs. Schäffer and Budenberg, a bent steel tube of elliptical section forms the pressure-measuring element, the movement of the free end of this tube being transmitted to a pencil movement of the Richards or Thompson type, and a record being obtained of the variation of pressure in the apparatus to which it is attached. Pressures as high as 10 tons per square inch are thus readily dealt with.

Indicators for Very High Pressures.—For excessively high pressures, indicators having exceptionally small pistons are occasionally used. In a special instrument made by the Crosby Steam Gage Company for testing the action of pneumatic gun carriages, and for hydraulic work, etc., the usual piston has an extension in the form of a plunger 0·18in. in diameter, corresponding to an area of 0·025sq. in., or $\frac{1}{20}$ of the area of the ordinary piston.

The pencil motion is of a similarly heavy pattern to that used in the same makers' gas engine indicator, and in addition, the bracket of the instrument carries a vertical post against which the pencil arm bears lightly, when needed to prevent the pencil point being thrown away from the paper by very sudden shocks.

A bye-pass is formed in the metal at the side of the small cylinder, by which the pressure, when not too high for the capacity of the spring, can be admitted to the larger piston. When the small piston is to be used, this bye-pass is tightly closed, the pressures being read off on the ordinary scale, and the results multiplied by 20.

CHAPTER IV.

ERRORS OF THE INDICATOR.

HAVING described in some detail the various forms of modern indicators, we shall now consider the various errors inherent in the instruments and incurred in their operation.

Piston Friction.—A considerable amount of error is in some cases traceable to excessive friction of the indicator piston, this being often caused by grit carried into the cylinder from the pipes and other connections. Frequently this condition of affairs may be detected by placing the

FIG. 37.

finger lightly upon the upper end of the piston rod. The remedy in this case is obvious. Occasionally a new instrument will be found to exhibit an unusual want of freedom, in which case, after it has been thoroughly cleaned and lubricated, it may be connected to the engine cylinder and allowed to run for some time, the pencil mechanism having first been disconnected. An abnormal amount of piston friction frequently exhibits itself in the diagram by a series of step-like serrations in the expansion line as shown in Fig. 37, the horizontal portions of the steps

indicating a sticking of the piston at each of those positions in its descent.

When a strong spring is used, the friction caused by a tightly-fitting *clean* piston may not reveal itself in this way. If, however, the resulting diagram is compared with one taken under precisely similar conditions, but with a freely-moving piston, it will be found that the various events in the stroke are shown as occurring unduly late, owing to the sluggish response of the piston to the variations of steam pressure. A vertical admission line with the freely-moving piston may appear so distinctly sloping in the other diagram as to suggest insufficiency of lead, while the initial pressure attained will be less and the back pressure greater in the latter case than in the former. With a fairly early cut-off, however, the area of the two diagrams may not greatly differ, as the first-mentioned losses are compensated by the delayed formation of the expansion line; but the diagram taken with the tight-fitting piston may be very misleading as an indication of the action of the valve gear.

To determine the amount of the piston friction, the instrument is first allowed to make a few working strokes, after which the pencil lever is pressed down by hand, so as to slightly extend the spring, and then allowed to return to rest, the indicator being rapped with a small wood stick. When at rest, a horizontal line is drawn on the diagram. The pencil lever is next raised slightly, again allowed to come to rest, and another line drawn as before. The interval between these two lines is a measure of the sum of the *total* frictional resistances in both directions, and, assuming the pencil mechanism to be in order, almost the whole of the error so measured is attributable to piston friction. In some tests of one of the best forms of modern indicators, the mean amount of double friction with various springs was found to be as follows:—

Scale of spring (lb.)...	20	36	40	60
Double friction (lb.)...	0·6201...	0·6342...	0·6763...	1·008

It is scarcely necessary to remark that careful attention to the lubrication of the piston will conduce to smooth running, and prevent any tendency to stick or bind. Clean cylinder oil, castor oil, or valvoline will be found a much

better piston lubricant than the thin oil used for the joints of the pencil movement. The amount of lubricant required will depend upon the character of the work in hand, frequent attention being necessary when indicating gas, oil, or petrol engines.

Springs.—The chief error met with in this important detail is a want of regularity in compression under uniform increments of pressure, any inaccuracy in this respect being magnified in the diagram four, five, or six times, according to the type of pencil movement used. When it is considered that in the Crosby instrument, for example, an error of 0·0028in. in the compression of a 60lb. spring will produce an error of 1lb. in the diagram, the importance of accurate springs will be readily admitted.

From a number of tests of springs which have been made from time to time it would appear that the character and extent of this error varies considerably in instruments by different makers. The general tendency, however, is for the spring to give a continual gain in the pressure indicated, owing to the spring becoming gradually weaker as it is compressed. The amount of this error in new springs by the best makers does not usually exceed 3 per cent. when tested under steam, but after having been some time in use the error may be materially increased, especially if emery cloth has been generously used to remove rust from the coils. It is undoubtedly very desirable to periodically test the springs, and for exact work this involves the use of a standard mercury column. If an accurate steam gauge is available, an approximate test may be made by connecting a short horizontal length of piping to the boiler or main steampipe, as most convenient, fitting a regulating valve at the supply end of this pipe, while the outlet end is closed by a small cock, by which water of condensation may be allowed to escape. Two tee-pieces are inserted in the length of piping, in one of which the indicator is mounted, while to the other the steam gauge is attached by means of the usual syphon. Steam is first allowed to blow freely through the pipe to expel water and to thoroughly warm up the instruments. Then by means of the two cocks the pressure is regulated until some definite amount—10lb., for example —is indicated by the gauge. The paper drum of the

indicator is then rotated and a line drawn. This process is repeated at various intervals of pressure (which for springs up to 40lb. may be of 5lb. each, and for those above, 10lb.), a similar series of lines being drawn as the piston descends under successive reductions of pressure. The regulating valve is then closed, the small cock opened fully, and the atmospheric line drawn. The difference between any pair of corresponding lines is, as already pointed out, a measure of the friction of the piston, etc., at that point, and lines drawn midway between each pair will give a scale which may be regarded as the scale of the spring tested.

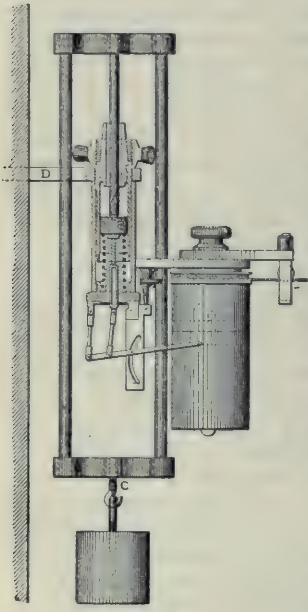

FIG. 38.

It will be seen that this method of calibrating the spring covers any errors in the area of the piston or in the pencil movement. It does not, therefore, give an absolutely true *spring* scale, but rather a true scale of the combination, which is of far more practical utility. If, however, the spring is used in another instrument, the scale will of course require to be redetermined.

A standard gauge not being always available, various devices for testing indicator springs by weights have been employed. One method used by Bollinckx is shown in Fig. 38. In this case the instrument is inverted, and is held in an exactly vertical position in the manner indicated, D being a bracket fixed to a wall or other convenient support. A light frame carries a rod A, the point of which bears upon the centre of the piston, as shown, while to the lower end of the frame weights may be attached, these being so selected that each represents a definite increase of

pressure per square inch upon the piston. It will be seen that in this method of testing, errors in the pencil movement will be covered, but any inaccuracy in the piston area will not be taken into account. The piston diameter (when hot) must therefore be ascertained, and the scale of the spring corrected accordingly.

The apparatus shown is quite suitable for outside-spring indicators. For enclosed-spring instruments it would not be difficult to arrange a method of jacketing the indicator cylinder with steam, the instrument being heated up to the working temperature (taken as 212° F.), since the strength of springs at this temperature is from $2\frac{1}{2}$ to 3 per cent. less than at 60° F.

Pencil Movement.—This may be briefly examined with respect to (*a*) Parallelism of the motion as influencing the accuracy of the vertical movement of the pencil; (*b*) Uniformity of ratio of pencil and piston motion; (*c*) Amount and distribution of the weight; and (*d*) Friction and effect of wear.

Commencing with the *Richards* pencil movement, it may be said that the greatest deviation of the pencil from the straight line occurs when the centre or pencil link is vertical. When this occurs in the lower part of the stroke, the pencil is drawn slightly too far from the line of motion of the piston rod, the deviation being similar in amount, but in the opposite direction, when it occurs in the upper part of the stroke. Assuming the path of the pencil to be divided into four equal parts, the two centre portions would satisfy every requirement of practical exactness; beyond this, in either direction, the error increases slightly until the maximum is reached, as already indicated, after which the deviation becomes less, vanishing at about the extreme limits of movement. As a result of this, the ratio of the movement of the piston and pencil is not maintained exactly, the scale being slightly more open in the centre than in the outer portions of the pencil travel.

The investigation of the effect of the weight of the pencil movement resolves itself into the determination of the resulting pressure upon the upper end of the piston rod. In the Richards instrument, the effect of the two main levers of the parallel motion will be found by multiplying

their weight by the distance of their centres of gravity from the fulcrums, and dividing this by the distance between the point at which the piston rod is connected and the nearest fulcrum. The effect of the pencil link is, however, much more serious, since this is to be multiplied directly by four, the ratio of the pencil motion to that of the piston in this instrument. In one indicator the weight of the two main levers, referred to the piston rod, was found to be 0·0242lb., while the weight of the centre link, similarly resolved, was 0·0548lb.

In order that the pencil movement may be compactly arranged, the small link connecting the piston rod to the parallel motion is made as short as conveniently possible. This implies a fair amount of movement of the link, and the consequent friction and wear at its two ends require to be well provided for, since any lost motion here would produce a fourfold effect upon the movement of the pencil. The pressure of the pencil on the paper also tends to increase the friction somewhat.

The *Thompson* pencil mechanism gives a close approximation to perfect rectilinear motion throughout the greater part of its movement: while the use of a comparatively long link to connect the piston and pencil lever ensures the motion of the pencil bearing practically a constant ratio to that of the piston.

It will be observed that although the original Thompson movement has the same number of joints as the Richards, three of them have a very small angular movement, and the friction and wear are therefore correspondingly less. Moreover, the arrangement is very "stable," owing to the comparatively long rods employed.

The *Tabor* pencil movement is essentially a modification of the Thompson arrangement, a point in the pencil lever being guided by a curved slot in place of being attached to a radius rod. In this way the effective weight of the mechanism is still further reduced, while a more exact rectilinear movement of the pencil is secured. The curvature given to the slot is such that the ratio of movement of piston and pencil is constant. In this case the number of joints is reduced to four, but there is to be added the friction of the roller in the slot, which, however,

tests have shown to be very small. It will be noted that as it is impossible for the small roller to be simultaneously in contact with both sides of the slot, the pencil lever must have a small amount of lateral play, but in this well-made instrument this is reduced to the smallest possible amount. It will be noted, moreover, that in this instance the effect upon the position of the pencil is not increased in amount by the magnifying action of the mechanism.

The *Darke* pencil movement (Fig. 15) gives a motion which is theoretically perfect, but although fewer actual joints are employed than in any other movement, the accuracy may be seriously impaired by wear of the sliding socket, or sleeve at the top of the piston rod, of the small carriage carrying the pencil, and of the latter in the vertical slot guide. Any lateral wear of the piston rod in the plane of the motion will also affect the accuracy. The makers have, however, provided against these sources of error by good workmanship and the provision of large bearing surfaces. Obviously everything depends upon the correctness of the position of the vertical guide plate, and this should be tested from time to time, as explained in Chapter IX.

During the upward stroke the friction of this pencil movement is somewhat greater in the lower and upper parts of its range, reaching a minimum in the centre of its travel; in the downward movement the friction is less marked. Careful lubrication of the sliding surfaces will, however, reduce this effect very considerably.

The *Crosby* pencil movement involves the use of six joints, and of the four links employed the radius rod marked 15 in Fig. 16 moves through a fairly large angle. The mechanism gives very good results as regards both the rectilinear movement of the pencil and the constancy of multiplication of motion. The multiplying ratio in this instrument is six, and therefore any lost motion at the joints marked 12, 18, and 19, Fig. 16, would affect the accuracy of the diagram to a greater extent than in other instruments. The makers have duly recognised this, however, and have provided means for accurately adjusting the mechanism, while, further, the workmanship bestowed upon this instrument is of a very high class. The mechanism as a whole is very light, but it will be noted that it is not independent of the

piston rod, and if disconnected therefrom, it ceases to form a "stable linkage."

The *Dobbie-McInnes* pencil movement is in effect a Thompson movement, in which the piston rod is connected to the radius rod instead of acting directly upon the pencil lever as in the older instrument. Both the radius rod and connecting link are somewhat shorter than in the Thompson instrument, but in general the remarks made in connection with the latter apply.

Inertia of Piston and Pencil Movement.—In considering the effect of the inertia of the piston, piston rod, and the attached mechanism, it becomes necessary to determine for each separate part its "equivalent mass" at the piston. The weight of the piston and piston rod, together with that of the collar or fitting at the lower end of the spring, are the only parts which are taken into the computation with their weights simply, although the connecting link may generally be also so included without material error. The equivalent mass of the spring (without the end fitting) is one-third of its actual weight.

Space will not allow of a full investigation of this point, but it may be remarked that in a Richards instrument it was found that while the weight of the piston, piston rod, and connecting rod, together with the equivalent mass of a 20lb. spring, was 0·1053lb., the equivalent mass of the pencil mechanism was 0·2253lb., of which latter amount 0·166lb. was due to the pencil link alone.

The effect of the inertia of the moving parts in producing oscillation of the pencil has been very fully investigated by Prof. Osborne Reynolds.* He found that the resulting error, so far as the *area* of the diagram is concerned, depends upon the magnitude of the oscillations in some measure, but to a greater extent upon the smallness of the number which appears in each revolution. Their magnitude is affected by the degree of suddenness of the steam admission, and will therefore be different for different valve gears and engine speeds. The general effect of the impact of the steam upon the piston approaches that of a suddenly imposed load, and tends to force the piston through twice

* Proceedings Inst. Civil Engineers, vol. 83.

the distance that it ought to rise. A similar but much less marked action takes place at any sudden change of pressure. The oscillations thus induced are, however, modified and gradually destroyed by the friction of the piston, together with that of the pencil and attached mechanism.

It frequently results that undulations of considerable magnitude do not affect the area of the diagram to any very great extent; but the distortion of the figure thus produced quite destroys its value as an indication of the action of the steam. The minimum number of oscillations in a complete revolution, consistent with clearness, appears from Prof. Reynolds' experiments to be about thirty, and when the number approached fifteen, a fair diagram could only be obtained by exerting considerable pressure on the pencil. It is needless to remark that such a method of reducing the distortion of the diagram is quite inadmissible if the results are to have any pretension to accuracy.

It is obvious that a reduction of the inertia effect is to be obtained by diminishing the weight of the moving parts. This has been the constant endeavour of the makers of modern indicators, and further progress in this direction appears to be limited to the use of aluminium for the piston, piston rod, etc. It will be understood that it is always possible to diminish undulations by employing a stronger spring, but this entails a consequent reduction in the height of the diagram, which is often objectionable. It is true that this may in turn be remedied by increasing the multiplying leverage of the pencil movement, but for the reasons already cited this plan is not to be recommended. On the whole, the most successful method of obtaining a diagram free from undulations at very high speeds is to use some form of optical indicator.

Pencil Friction.—The general effect of the friction resulting from the pressure of the pencil on the paper, and that of the mechanism in sustaining it, is to increase the area of the diagram, since by always acting in opposition to the motion of the pencil it causes the steam and expansion lines to be drawn above, and the back pressure and compression lines below, their true positions. The extent of the error thus incurred depends in some degree upon the condition

of the surface of the paper and the kind of pencil used, but chiefly upon the pressure brought to bear upon the latter. This pressure should never exceed that necessary to draw a fine working line, since, in some instruments, even this produces an error of 0·5lb. in each direction. From the above considerations it will be apparent that the extent of the error from pencil friction will depend upon the cut-off of steam and other conditions. Thus, in the case of a locomotive notched up to mid-gear, it appears possible for this error to amount to 5 per cent.; while on slow-moving engines having a late cut-off and little compression, the effect will be very much less marked.

Errors in the Drum Motion.—The chief causes of errors in the motion of the paper-carrying drum are (*a*) the inertia of the drum, (*b*) the imperfect action of the spring, and (*c*) the friction of the drum. The general effect of these disturbing influences is to vary the tension of the driving cord, causing it to stretch unequally, and hence producing an irregular displacement of the diagram in the direction of its length. With a positive drum-driving arrangement these errors would be eliminated, but the practical difficulty of adopting such a method of actuating the drum appears to prevent its use under any but exceptional circumstances. Reducing gears, in which only a very small amount of cord is used, will be found described in Chapter VII., and with these the inaccuracies due to the drum action are reduced to a very small amount indeed.

It will be seen that the conditions governing the oscillations of the drum are very similar to those influencing the motion of the balance-wheel of a watch. Every drum, together with its attached spring, has its natural period of oscillation, which becomes longer as (1) the moment of inertia of the drum or (2) the length of the spring is increased. When the drum and spring are so arranged that the period of oscillation of the drum coincides with the period of movement of the engine crosshead, the tension on the driving cord, and consequently the amount of stretch, will remain uniform. On the other hand, if these two periods differ materially, the cord tension will be correspondingly affected. With speeds so low that the inertia of the drum is negligible, it is readily seen that the length of

the diagram will be diminished by an amount $d\,y$, d being the difference between the tensions of the drum spring at the extremities of the stroke in pounds, and y the stretch of the driving cord in inches per pound of tension. As the speed increases, the inertia of the drum gradually reduces this shortening of the diagram, until at the critical speed the one effect exactly neutralises the other. With still higher speeds the inertia effect preponderates, and the diagram becomes elongated. This action is modified by the action of the spring, and it may be shown that—since the effort of the drum spring is usually directly proportional to the angle through which the drum moves, and also since the accelerating force vanishes at mid-travel—if the difference of the drum-spring tensions at the ends of the stroke is twice that necessary to overcome the inertia of the drum, the cord will remain uniformly stretched.

All modern indicators are provided with means for adjusting the drum-spring tension, and this should be varied to suit the particular speed in each case. Since, however, increase of tension results in the ultimate stress on the cord being increased, only sufficient tension should be used as will ensure the cord being kept taut.

The general effect produced by the combined influences of the imperfect spring action and the inertia of the drum is to change the location of portions of the diagram, the direction and amount of such disturbance depending upon the consideration outlined in the foregoing; but with a suitable drum-spring tension the effect upon the mean effective pressure is very slight. Some indication of the tension required may be obtained by comparing the lengths of diagrams taken at low and full speeds. It may be noted, however, that when the driving cord or wire is kept taut by means of an auxiliary spring (described later), the drum-spring tension may be correspondingly decreased, leaving the sum of the tensions about equal to the drum-spring tension which would have been required ordinarily.

The error caused by friction of the drum is one which often affects low-speed diagrams appreciably. The extent of the error depends upon the lubrication of the drum spindle and the elasticity of the cord, and may therefore be reduced to a negligible amount by careful attention to the

first point, and by the substitution of wire or some positive form of driving gear for the cord ordinarily employed. The amount which the cord will stretch is evidently $f\ y$ inches, f being the tension in pounds required to overcome the frictional resistance of the drum, and y, as before, the yield of the cord in inches per pound of tension. In the forward stroke the drum will not commence to move until the cord has stretched by this amount; while in the return stroke the tension will be diminished and the length decreased to a similar extent. Thus the drum will be $f\ y$ inches behind its true position during the complete revolution, and the relative position of the two parts of the diagram will be displaced in consequence, that portion described during the return stroke being $2\ f\ y$ inches in advance of that described during the forward stroke. It will be readily seen that the effect of drum friction becomes more marked as the amount of expansion and compression is greater. Thus, in diagrams from locomotives, at or near mid-gear, and in gas and oil engines, the effect upon the mean effective pressure may be very appreciable.

As the errors of the drum motion mainly affect the accuracy of the diagram by the stretching of the cord, wire may often be substituted for the latter with advantage, or some positive method of driving the drum adopted. This point is further discussed in Chapter VII.

Other Sources of Error.—In the foregoing it has been assumed that the indicator is of rigid construction, and that no "springing" occurs by which the relative position of the steam cylinder and paper drum of the instrument is affected. This assumption is, however, not always warranted, as in the endeavour to reduce weight as far as possible, the arm or bracket carrying the drum spindle has in some cases been lightened sufficiently to allow the drum to spring slightly out of its true position under the sudden pull of the cord, this effect being of course much more marked at high speeds, and particularly when the cord is led directly away from the guide pulley on the instrument and in a vertical direction. It is readily evident that a constructional defect of this kind may seriously impair the truth of the diagram, and it is therefore fortunate that it is only very rarely met with.

The springing of the instrument bodily, owing to imperfect fixing or want of rigidity of the piping carrying the indicator, will affect the length, and consequently the accuracy, of the diagram, as will also the striking of the drum against the stops which limit its motion.

Some slight inaccuracy may be incurred owing to the expansion of the piston of the instrument, but for accurate tests the diameter should be carefully measured while hot. This error will be greater in indicating gas engines, owing to the very high temperatures reached. If diagrams are taken in a damp atmosphere and afterwards measured in a dry one, the shrinkage of the paper will reduce the area of the diagram, and the apparent power developed will appear correspondingly less than the true amount. The extent of this error will depend upon the thickness and character of the paper, but under usual conditions it will be negligible.

CHAPTER V.

THE ATTACHMENT OF THE INDICATOR.

AS a true delineation of the action of the steam in the cylinder can only be obtained when the pressure on the indicator piston varies in exact accord with that acting on the engine piston, it is obvious that a free and direct connection with the engine cylinder is of the first importance. From this it follows that to secure the best possible results the indicator should be attached directly to the engine cylinder, without any intervening piping, bends, cocks or valves, all of which tend to impede the flow of steam to and from the cylinder of the instrument, and hence to impair the accuracy of the diagram. For convenience in operating and other considerations, the instrument is preferably placed in a vertical position; hence it is evident that in horizontal engines the connection will be best effected by attaching the indicator directly to the upper part of the convex body of the cylinder. Most engine builders of repute make provision for such a connection by forming two bosses on the cylinder body, usually as projections from the flanges, and through these, holes are drilled directly into the clearance spaces. When not in use these holes are capped with hexagon or square-headed plugs of brass or gun-metal. If the engine is to be frequently indicated, it is preferable to permanently fix a pair of indicator cocks in the cylinder, these being covered with brass caps when the instruments have been removed.

If indicator bosses are not provided, it will be necessary to drill into the body of the cylinder, and in deciding upon the position of the holes in this case, regard should be paid to the arrangement of reducing gear which will be adopted.

As already stated, the best position for the instrument is on the top of the cylinder in horizontal engines, but if the conditions are such that this will involve the use of a guide pulley for the indicator cord, a more direct driving and better results generally will not infrequently be obtained by attaching the indicator to the side of the cylinder. In horizontal engines having valves of the Corliss type this will invariably be the position selected unless the cylinder covers are drilled. In either case bends will be required, and these should be of easy curvature, having a radius of from 4 to 6in. In no case should right-angled elbows be used.

Care must always be taken to locate the holes fairly in the centre of the clearance space at each end of the cylinder, and in such a position that they are not wholly or partially covered by the piston when it reaches the end of the stroke. In drilling the cylinder, situations directly exposed to the incoming steam must be carefully avoided; nor should the holes be drilled near the steam passages, as the current of steam passing over orifices so situated reduces the pressure in the indicator, and therefore vitiates the accuracy of the diagram.

When the cylinder has to be drilled, it will generally be found necessary to use a nipple and socket, or a short piece of pipe, in order to bring the cock clear of the lagging. Whenever possible, it is preferable to drill the holes with the cylinder covers removed, when, if necessary, a channel may be cut in the cover in order to give a free passage to the steam. When this is not convenient, care must be taken to prevent chips of metal from entering the cylinder. A slight admission of steam to the cylinder while the holes are being drilled may prove of service in this connection in the absence of any better expedient.

In vertical engines connection is usually made at the side of the cylinder by means of bends; but in many cases the cover is drilled and the instrument attached direct For the lower end, the side of the cylinder must be drilled in any case. In inverted-cylinder engines the lower cover should not be drilled, as water accumulates in the instrument, causing trouble in working.

The size of holes necessary is governed to some extent by the size of the engine. For the direct attachment of the

indicator to engines of small and moderate dimensions, it will suffice to drill holes $\frac{1}{2}$in. in diameter enlarging the outer portion to $\frac{5}{8}$in., this being the tapping size for the $\frac{3}{4}$in. Whitworth bolt thread usually found upon the cocks supplied with instruments of English make. In some indicators of American manufacture, however, the cock is screwed with a $\frac{1}{2}$in. pipe thread. If piping is used, it should not be less than $\frac{3}{4}$in. diameter in small engines; while in large marine engines it is often $1\frac{1}{2}$in. in diameter.

It is highly desirable that diagrams should be taken from both ends of the cylinder. For, in the first place, the diagram obtained from one end, however correctly it may exhibit the action of the steam during the one stroke, is useless as a measure of power unless the diagram from the opposite end is precisely similar. This, it need scarcely be observed, is of exceedingly rare occurrence. Moreover, however carefully the valves may be set, the indicator will frequently detect a difference in the steam distribution. Obstructions in the steam or exhaust ports and passages; defective action of the valve owing to wear in the actuating mechanism; unequal lead and cut-off, and several other influences, cause differences in the diagrams obtained from the two ends of the cylinder.

Not only should diagrams be taken from each end of the cylinder, but, wherever practicable, they should be so taken as nearly simultaneously as possible. This is of especial importance in engines fitted with automatic cut-off gear, and in any cases where the load varies rapidly. The most satisfactory means of accomplishing this is to use two instruments, each directly connected to the cylinder as previously explained, and operated by independent persons. In many instances, however, this method is inconvenient, and a pair of indicators are not always at command. Under these conditions, the single instrument must either be changed from one end to the other, or else placed in the centre of a length of piping connecting the instrument with either end of the cylinder at will, this being most conveniently effected by mounting the indicator in a three-way cock. As to which plan it is advisable to adopt, much depends upon the object in view. If a close examination of the action of the valve gear is aimed at, the direct

attachment will give the most reliable result. But for power-measurement purposes a marked change in the load may easily occur while the instrument is being transposed, and therefore the pipe connection will be generally the most serviceable.

It is undoubtedly true that lengths of piping, bends, and cocks exert a more or less marked influence upon the transmission of pressure to the indicator piston, and the direct method of connection has the advantage of reducing this error to the least possible amount. But if, as already suggested, the plan involves the risk of much greater errors owing to changes in the load on the engine, it is evident that piping to a three-way cock offers the best solution of the difficulty. In indicating locomotives this latter arrangement is almost invariably adopted.

In arranging systems of pipe connections all unnecessary bends should be studiously avoided, as also abrupt changes in the direction of steam flow. Bends of easy curvature should be employed, and the pipes arranged to be self-draining whenever possible. When this latter condition cannot be fulfilled, drain cocks are frequently required to be fitted to the pipes, especially in large engines. Not infrequently shut-off cocks are provided near the connections to the cylinder, so that the pipes are not always subjected to the steam pressure. In large engines, asbestos-packed cocks are used, these being preferably attached to the cylinder by flanged joints. In smaller engines screwed unions are more generally adopted, and not infrequently angle valves are employed. This is not good practice, as the sudden change in direction of the steam, together with the generally impeding effect of the valve, has often a marked influence upon the steam passing through it. In any case a three-way cock should always be used in addition, when shut-off cocks are provided near the cylinder. This precaution is neglected in an arrangment of piping very frequently adopted, in which two-angle or other valves are used close to the cylinder, their outlets being connected by a length of piping (Fig. 39), midway in the length of which is a tee carrying the indicator. A little consideration will show that with this arrangement it is impossible for the steam to exert that instantaneous effect upon the indicator

piston which is indispensable if accurate results are to be obtained. At the commencement of the stroke the steam rushing through the open cock must first fill the length of piping beyond the tee before its influence is fully felt by the indicator piston, and, as a result, the steam admission will be shown unduly late. The subsequent action of the steam will also be represented more or less incorrectly, as the changes of pressure which occur will be somewhat modified by the presence of the volume of steam beyond the indicator. This effect will of course be more marked when long lengths of pipe are employed.

If the two stop valves or cocks are placed close to the tee,

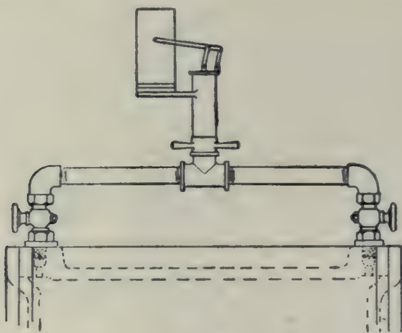

FIG. 39.

a much better arrangement is obtained, as practically all the superfluous length of pipe is then cut out. The abrupt change of direction due to the tee remains, however, as a decidedly objectionable feature, and the substitution of a three-way cock for the two valves and tee-piece is certainly to be recommended. The latter arrangement, with a pair of easy bends to the cylinder, is probably as good a system of piping as can be devised; but if the arrangement is intended to be permanent, it is advisable to fit straightway shut-off cocks as close to the cylinder as possible, not only to relieve the piping of the steam pressure, but also to obviate the unnecessary addition to the clearance space which the pipe provides. The latter item is in general only of small

moment, but as it is recognised good practice to reduce clearance volume and surface to the least possible amount, the point is one which is worthy of consideration when arranging a permanent system of indicator piping.

When the piping is at the side of the cylinder, the same arrangements are available, except that the bends connected to the cylinder will be in a horizontal, and the three-way cock in a vertical position. If the cylinder covers are drilled, two bends at each end will be needed, coupled by means of a nipple or a short piece of pipe, as may be necessary. Piping arranged in this way is less rigid than the more direct connection, and if the cylinder is fairly long and the pressure high, trouble may be experienced owing to the springing of the pipe. If possible, the piping should be additionally supported by securing it to the cylinder by some convenient means, since, as previously pointed out, any movement of the indicator due to the springing of the piping introduces inaccuracies. Rigid support of the indicator is sometimes secured by mounting it in a special fitting secured to the cylinder or steamchest, as most convenient. An instance of this is given in Fig. 55 (page 93), which shows the method of piping adopted for indicating an American locomotive. As will be seen, the piping, although fairly long, is tolerably direct, while a very secure fixing is obtained for the indicator by mounting it in the three-way fitting bolted to the steamchest cover.

In the case of piping for vertical cylinders it is difficult to avoid the use of a tee and a bend if the indicator is to be placed in a vertical position and connected to the pipe at the centre of its length. Shut-off cocks must, of course, be used in this arrangement, and these, as previously stated, should be placed close to the tee-piece. Special three-way cocks are occasionally employed, however, and in these the outer branch, in which the instrument is mounted, forms the required bend. In an arrangement at one time commonly adopted, the indicator was mounted in the upper branch of an ordinary three-way cock attached directly to the upper end of the cylinder, a length of piping being used to connect the lower branch of the cock to the lower end of the cylinder. With such a system of piping, inaccuracies would be introduced in the diagram from the

lower end of the cylinder owing to the retarding influence of the intervening length of piping. In cases where no other arrangement than this is admissible, care should be taken to provide a pipe of ample size and to cover it carefully with non-conducting material so as to prevent condensation of the steam as far as possible. It is scarcely necessary to observe that this latter precaution should always be taken in the case of piping for locomotive cylinders, and any others which are in exposed situations.

Iron piping being inexpensive and readily available, is very largely used for indicator connections, but brass or copper tubing is much to be preferred. If iron piping is used, care should be taken to remove the loose scale frequently found inside such tubing, bends, etc. A slight hammering of the tube will generally be found to detach a quantity of scale, which otherwise might find its way into the indicator cylinder, causing an incalculable amount of injury to the instrument. Burrs should be removed from the ends of all pipes, and in making the joints no red lead putty or similar material should be used, a little wicking or cotton waste being wound in the pipe threads if required. When the pipes have not been used for some time they should be blown out thoroughly before attaching the indicator, so as to dislodge any rust which may have formed in them.

Copper and brass piping being free from the foregoing objections, is much more suitable for this purpose, and if the fitting is intended to be permanently attached to the cylinder, the consideration of improved appearance should not be overlooked. Fittings of this kind as supplied by some indicator makers have a simple form of expansion joint provided in connection with the three-way cock (Fig. 41). This is a decided advantage, especially if the pipe is long.

Since, as already mentioned, the indicator piping usually has to also form a support for the instrument, all joints must be securely made and the piping rendered perfectly rigid. Stop valves and other fittings used should be tested as to steam-tightness, *particular care being taken with the three-way cock*, if one is used, as any leakage of this fitting, or of the usual indicator cock, is calculated to displace the atmospheric line and otherwise introduce inaccuracies.

In Fig. 40, representing the three-way cock supplied by the Crosby Valve and Engineering Company, the expansion joint above referred to is shown on the right of the sectional view. It will be seen that the steamway through the plug is large and direct. As shown, the left-hand pipe is communicating with the indicator outlet, while turning the

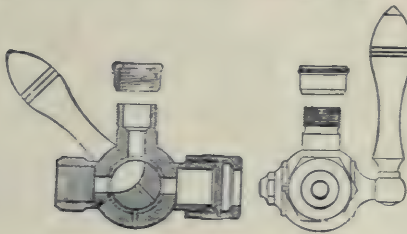

FIG. 40.

handle through an angle of 90° places the instrument in connection with the other end of the cylinder. When the handle is in the vertical position, communication with the cylinder is entirely cut off; but the small hole in the plug then agrees with the hole at the bottom of the shell of the cock and admits atmospheric pressure to the

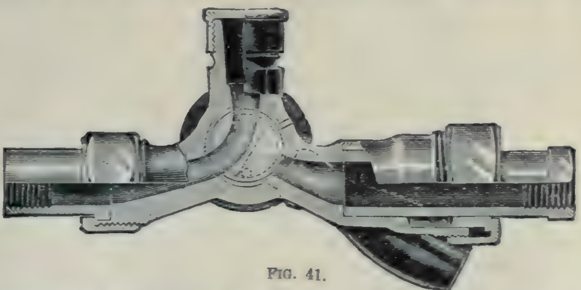

FIG. 41.

underside of the indicator piston, while also serving the very useful purpose of draining the instrument of water. With a cock so arranged, the indicator may be directly connected to the fitting without using the usual cock supplied with the instrument. Fig. 41 is a sectional view of the three-way cock supplied by the makers of the Tabor indicator.

CHAPTER VI.

ERRORS OF INDICATOR CONNECTIONS.

THE precautions which should be observed in connecting the indicator to the engine cylinder having been alluded to in the preceding section, it now remains to briefly discuss the nature and extent of the errors which may be introduced by faulty methods of attachment.

Errors due to Faulty Connections.—The error introduced by the partial or complete closure of the indicator orifices, owing to the piston covering them at the end of the stroke, may often reach most serious proportions. It generally

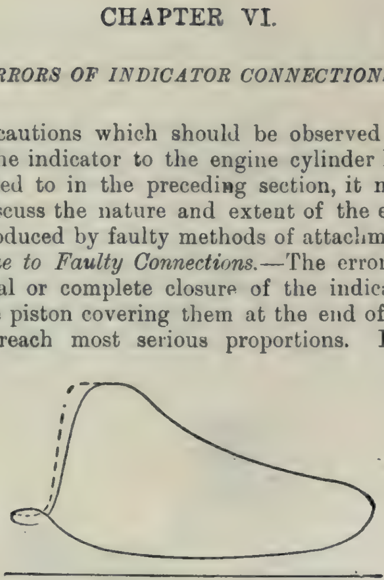

FIG. 42.

results in the formation of a loop as shown in Fig. 42, the fall of pressure thus indicated being due to the leakage, past the piston of the instrument, of a portion of the small volume of confined steam. With higher speeds or long connecting pipes the loop may appear flattened, as shown by the broken lines; leakage past the engine piston is, however, not without effect upon the shape of the loop. Fortunately, this condition of affairs is readily discovered by inspecting the diagram, and with the careful operator

there is little liability of errors due to this cause escaping detection. The partial choking of the cylinder orifices, owing to red lead and other foreign substances obstructing the passages, is, however, not so readily revealed in the diagram, and therefore too much care cannot be taken to ensure the instrument being always in free communication with the cylinder. Indicator cocks permanently fixed in the cylinder should receive periodic examination, as with many kinds of cylinder oil a pitch-like substance is deposited in such fittings, which may ultimately diminish the opening to a very considerable extent.

Loss of pressure is caused by interposing a right-angled elbow between the cylinder and the instrument, by the use of tee pieces, globe valves, etc., but it is impossible to estimate the extent of the error thus incurred. In exposed situations a similar loss may result owing to condensation of the small volume of steam contained in the indicator cylinder. This will be particularly the case in indicating locomotives running at high speeds, under which circumstances it is desirable to protect the cylinder of the instrument, as well as the piping, by a temporary lagging. The method adopted in the Dobbie-McInnes indicator of sheathing the cylinder with vulcanite is of decided value in this connection. Such precautions as these are often held to be unnecessary, but it is to be remembered that the behaviour of the inconsiderable volume of steam contained in the cylinder of the instrument is assumed to be identical with that of the main body of steam in the engine cylinder, an assumption only warranted when the sample steam is used under precisely similar conditions.

Water in the Cylinder, etc.—The presence of water in the cylinder, due either to priming or condensation in the instrument and connected piping, is fatal to accuracy in the diagram. It frequently results in the production of curious figures, mainly characterised by undulations in the steam and expansion lines. It is scarcely necessary to remark that it is useless to attempt to take a card until the pipes and instrument are entirely cleared of water.

Errors due to Pipe Connections.—The effect of long pipe connections upon the accuracy of the diagram has been a much-discussed topic during recent years, and many experi

ments have been made in order to ascertain the nature and extent of the error under various conditions. Tests made many years ago with instruments having heavy reciprocating parts gave deficiencies in the area of the diagram of from fifteen to forty per cent., but recent experiments have shown that not only is the error generally of far less extent, but also that it is not always manifested in the same manner —the diagram area in some cases being increased and in others diminished by the interposition of pipe connections— as compared with cards taken with direct connection under otherwise precisely similar conditions. It would appear that an explanation of these diverse results is to be found in the fact that in general those diagrams which give an augmented area are those taken with early cut-offs, while as the ratio of expansion becomes less, the error diminishes, and in some cases may change into one of an opposite

FIG. 43.

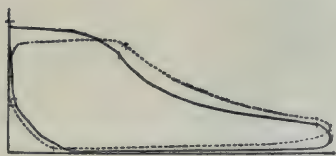

FIG. 44.

character. Some support is lent to this view by a consideration of the diagrams shown in Figs. 43 and 44, which were taken under similar conditions, except as regards the cut-off. It is seen that sluggish action of the steam with the pipe connection (shown by the dotted lines in both cases) has the effect of adding considerably to the area in the case of Fig. 43 (one-eighth cut-off), while in Fig. 44 (one-third cut-off) the added area at the expansion line is almost compensated for by the deficiency shown near the steam and exhaust lines. As a matter of fact, the dotted diagram in Fig. 43 shows an excess of 35·3 per cent. in the mean effective pressure, as compared with the diagram taken with the direct connection, while in Fig. 44 the excess is only 3·4 per cent. In these tests the initial pressure was 80lb. per square inch, the length of pipe 10ft., and the number of revolutions 200 per minute.

While it is impossible to draw general conclusions from isolated tests, it would appear that the above suggestion offers an explanation of the fact that in a few instances careful tests have not shown any appreciable difference between the results obtained by the two methods of connection so far as the *area* of the diagram is concerned. This, however, is largely a matter of coincidence, and may be taken as the exception which proves the rule; for certainly the weight of evidence is confirmatory of the generally accepted view that pipe connections almost invariably impair the accuracy of the diagram to a greater or less extent.

Prof. W. F. M. Goss, from whose paper in the Transactions of the American Society of Mechanical Engineers the foregoing diagrams are taken, has given the following summary of the results obtained from an extensive series of tests:—(1.) If an indicator is to be relied upon to give a true record of the varying pressures and volumes within an engine cylinder, its connection therewith must be direct and very short. (2.) Any pipe connection between an indicator and an engine cylinder is likely to affect the action of the indicator; under ordinary conditions of speed and pressure, a very short length of pipe may produce a measurable effect in the diagram, and a length of 3ft. or more may be sufficient to render the cards valueless except for rough or approximate work. (3.) In general, the effect of the pipe is to retard the pencil action of the indicator attached to it. (4.) Other conditions being equal, the effects produced by a pipe between an indicator and an engine cylinder become more pronounced as the speed of the engine is increased. (5.) Modifications in the form of the diagram resulting from the presence of a pipe are proportionally greater for short cut-off cards than for those of longer cut-off, other things being equal. (6.) Events of the stroke (cut-off, release, beginning of compression) are recorded by an indicator attached to a pipe, later than the actual occurrence of the events in the cylinder. (7.) As recorded by an indicator attached to a pipe, pressures during the greater part of expansion are higher, and during compression are lower, than the actual pressures existing in the cylinder. (8.) The area of diagrams made by an indicator attached to a pipe

may be greater or less than the area of the true card, depending upon the length of the pipe; for lengths such as are ordinarily used, the area of the pipe cards will be greater than that of the true cards. (9.) Within limits, the indicated power of the engine is increased by increasing the length of the indicator pipe. (10.) Conclusions concerning the character of the expansion or compression curves, or concerning changes in the quality of the mixture in the cylinder during expansion or compression, are unreliable when based upon cards obtained from indicators attached to the cylinder through the medium of a pipe, even though the pipe is short.

While agreeing in general with the foregoing conclusions, we are of opinion that the error introduced by the use of well-arranged pipes of 3ft. in length will not usually be sufficient to impair the practical accuracy of the diagram.

CHAPTER VII.

INDICATOR REDUCING GEAR.

IT has been pointed out in a previous chapter that in order to obtain a correct record of the action of the steam, the motion given to the cord which actuates the paper-carrying drum of the indicator must be an exact reproduction (although on a reduced scale) of the motion of the engine piston or crosshead. It is therefore necessary to devise a convenient means by which the length of the stroke of the engine can be reduced to that required for the indicator diagram, this being usually from 2 to 5in. long, according to circumstances. As this can be effected in a variety of ways, it is not surprising to find that a very large number of different arrangements are in use, many of which, however, give results which are far from accurate. It is probably not too much to say that greater errors are introduced by the use of imperfect reducing gear than any other cause, and it is therefore highly desirable that more attention should be given to this matter than is commonly assumed to be necessary.*

For descriptive purposes, the various arrangements of reducing gear employed may be very conveniently divided into four classes: (1) Pendulum Levers, (2) Pantagraph Motions, (3) Reducing Wheels, and (4) Miscellaneous Devices.

Pendulum Lever Reducing Gear.—Of this, the most commonly adopted type of reducing gear, many forms are met with. The most simple form is that shown in Fig. 45, which, as will be seen, consists of a slotted pendulum lever

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* The comparative accuracy of indicator reducing gears is discussed in Chapter VIII.

arranged to turn about a fixed pin C at its upper end, while the slot at the lower end engages with a pin A which is fixed in the crosshead of the engine. For temporary use the lever may consist of a strip of straight-grained deal about 1 or $1\frac{1}{4}$ in. in thickness, and tapering in width from about 5in. at the upper end to about 3in. at the lower extremity. The upper centre or pivot on which the lever turns may consist in this case of a coach-screw, which, with large washers between the screw head and lever, and also between the lever and the support, provides a fairly satisfactory centre. The slot in the end of the lever must be "square" with the face of the lever, parallel throughout its length, exactly in the line of centres or axis of the lever, and fitted to the pin with which it engages, so as to avoid lost motion or "play." The length of lever should not be less than $1\frac{1}{2}$ times the length of stroke, and the fixed centre on which it swings should be located vertically over the centre of the path described by the pin in the crosshead. This form of lever may be rendered more durable by attaching to each side of the slotted end thin iron plates in which the slot is made. The hole in the upper end of the lever may also be bushed with a piece of brass tubing. The pin P, to which the looped end of the cord is attached, may consist of an ordinary wood screw, of which only the neck is allowed to project from the face of the lever. This

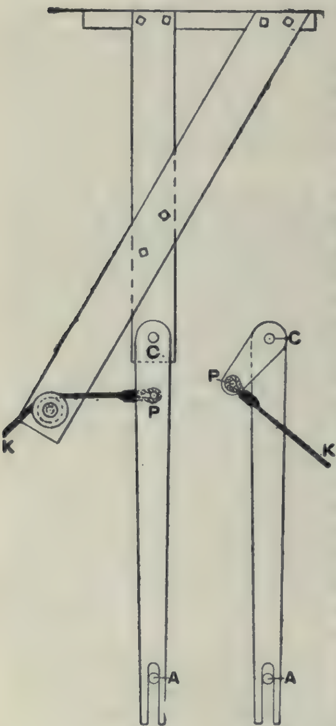

FIG. 45. FIG. 46.

must, of course, be set in the centre line of the lever, and at such a distance below C as will give the desired motion to the cord.

In order to determine the position of the pin P, it should be noted that the length C P must bear the same proportion to C A as the desired length of diagram bears to the piston stroke. From this it follows that the rule to find the distance of P from C is:—Multiply the length of the lever in inches by the desired length of diagram, and divide the product by the stroke of the engine, also in inches. Thus, if a travel of 4in. is required to be given to the drum of the indicator when a 36in. lever is used and the length of stroke is 24in., we have

$$\frac{36 \times 4}{24} = 6\text{in.}$$

as the distance of the centre of P from the centre of the fixed fulcrum C.

For a temporary arrangement such as that above described, the fixed centre may often be carried by a stout piece of timber depending from and fixed to a beam in the engine-room; but in order to secure the necessary rigidity, a brace will generally be required, as shown in Fig. 45. Occasionally it may be possible to secure the required fixing by wedging a wood post between the engine bed on the floor and some overhead support; but the precise arrangement adopted will depend upon circumstances, and to a great extent it must be left to the ingenuity of the operator to make the best use of the means available.

For permanent use, a more substantial form of forged lever may be employed, the dimensions of which will vary with the size of the engine. For a stroke of 24in. it may consist of a flat iron bar not less than 36in. long, $\frac{1}{4}$ or $\frac{3}{8}$in. thick, and tapering in width from 2 to $1\frac{1}{4}$in. In this case the fixed centre will be best carried by a light, rigid standard fixed to the guide bars or other convenient part of the engine, and in such a manner as will allow of it being readily removed when desired. A turned standard or upright bolted to the upper guide, and a lever forged from a round bar and finished bright, forms a very neat arrangement of permanent reducing gear.

The method adopted for fixing the pin A will depend upon the arrangement used for guiding the crosshead, and also upon the construction of the latter. In many cases the side of the crosshead is accessible, and a hole may be drilled in any convenient position to receive a screwed stud of sufficient length to project about 1in. through the slot in the lever. Frequently the set-screws or nuts provided for the adjustment of the crosshead may be pressed into service, in which case a foot may be forged on the end of the stud, in which holes may be drilled as convenient. Where the crosshead is guided in such a manner that the side attachment is inadmissible, the upper surface may sometimes be drilled and tapped for two small set-screws, passing through holes in the end of the stud, which is flattened for the purpose. Fig. 72 (page 105) may also be suggestive in this connection.

With this reducing gear—and in fact with any arrangement in which the cord is attached to a pin set in the axial line of a lever—special care must be taken to lead the cord away in a direction parallel to the line of stroke of the engine. Inattention to this point will result in the production of diagrams which are utterly worthless. Of course, by the aid of guide pulleys, the cord may afterwards be led in any direction to the instrument. For these, however, additional fixings will be required which are often difficult to obtain; besides which, guide pulleys are more or less a source of trouble, especially at high speeds, and their use is therefore to be avoided whenever possible. To this end, modifications of the simple pendulum lever have been adopted, one of which is shown in Fig. 46. Here a short lever C P, carrying the pin for the cord, is securely fixed to the main lever C A, and in such a position relatively thereto that when C A is vertical (corresponding to the centre of the piston travel) the line C P will be at right angles or "square" with the direction of the cord P K. In this way the cord may be led directly to the guide pulleys on the indicator without distorting the diagram. It will be evident that if two indicators are used, or if one instrument is alternately used at each end of the cylinder, the direction of the cord will vary somewhat, and the position of the short lever will require to be slightly altered. With a fair length of cord, however, no very serious error will

result if the short lever is placed midway between the two positions which would otherwise be necessary. In place of a separate lever, a projecting piece may be attached to the main lever, in order to carry the pin P, the relative position of the imaginary line P C and the direction of the cord being the same as in the previous arrangement. The method shown in Fig. 46 is preferable when the short lever is to be placed some distance away from the main lever, as, for example, in Fig. 54, and in such cases a distance piece or sleeve may often be improvised by the aid of a piece of iron gas tubing, screwed for back nuts at each end. In this way the cord may be led direct to the indicator, as before described, and also *in a plane parallel to that in which the crosshead moves*—a requirement of the first importance with nearly all reducing gears, but which, it will be understood,

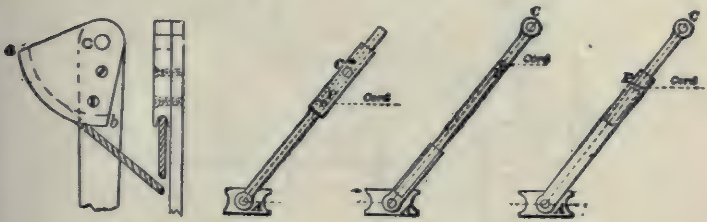

FIG. 47. FIG. 48. FIG. 49. FIG. 50.

does not apply *after* the cord has passed a guide pulley on its way to the indicator.

Another modification of the simple pendulum lever consists in attaching a circular segment to the upper end, as shown in Fig. 47. The arc *a b* is struck from the centre C on which the lever swings, and is of a radius sufficient to give the required movement to the cord, which latter lies in a groove on the edge of the segment, being secured to the latter at *a*. It will be seen that with the "brumbo pulley," as this segment is sometimes termed, the direction of the cord may be varied considerably without affecting the motion. As will be shown subsequently, however, the addition of the brumbo does not in most cases enhance the accuracy of the reducing gear. It will be understood that the segment may be fixed at any angle relative to the main

lever, and that it may either be attached directly to it or to a tubular distance piece or sleeve as previously described.

Disguised forms of the arrangement shown in Fig. 45 are not infrequently met with in the shape of telescopic reducing gears such as those shown in Figs. 48 and 49. In the first of these the cord is attached to the lower end of a tube, turning about a fixed centre or trunnion C. The pendulum takes the form of a round rod connected to the crosshead by the pin A. The arrangement shown in Fig. 49 is precisely similar in character, but more readily constructed. In the telescopic gear shown in Fig. 50 the tubular portion is made of sufficient length to carry the pin P to which the cord is attached. Fortunately this gear is

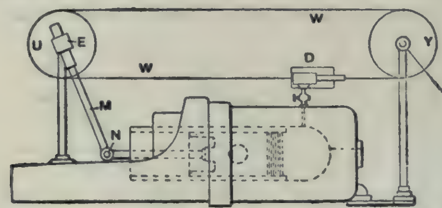

FIG. 51.

rarely met with, for it gives grossly inaccurate results, and should therefore never be employed under any circumstances.

The telescopic gear shown in Fig. 51, applied to a gas engine,* is identical in principle with the simple gear formed by combining Figs. 47 and 48. In this case, however, the required movement is given to the drum by attaching a fine steel wire W to its extremities and passing it round the two light pulleys U and Y, to the former of which it is secured. The latter can be moved outwards slightly, so as to maintain a suitable degree of tension on the wire. The pulley U is mounted on a horizontal shaft, which also carries the sleeve E, receiving the telescopic rod M. The lower end of the latter is jointed to a bar N, which is fixed in, and virtually forms a part of, the piston. In this way

* First Report to the Gas Engine Research Committee of the Institution of Mechanical Engineers.

the slide is positively driven in both directions, thus dispensing with the usual spring for producing the return movement. Although this forms a very convenient gear, it does not give accurate results, as a reference to diagram B¹, Fig. 80, will show.

A more accurate reducing motion may be arranged by reversing the driving arrangement of Fig. 45, fixing a pin B (Fig. 52) in the end of the pendulum lever A to engage

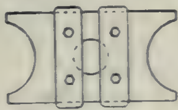

FIG. 53.

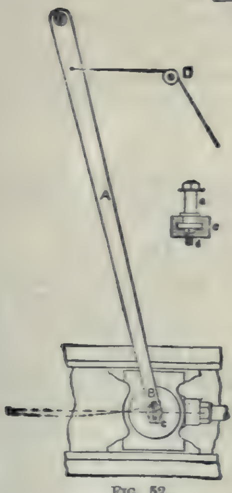

FIG. 52.

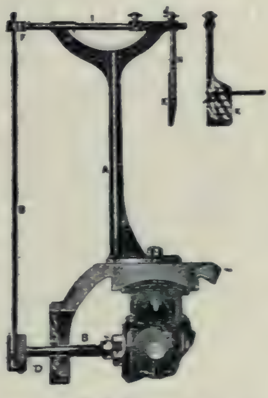

FIG. 54.

with a vertical slot C in a plate attached to the crosshead. A more durable arrangement is shown in the small view, in which the pin *a*, bolted to the lever A, has a large head sliding in the T-slot in the piece *c* as shown. The latter is fixed to the crosshead by two screws, one of which is shown at *d*.

As a substitute for the slotted plate, two iron strips may be fixed to the crosshead (Fig. 53); but however it is

arranged, care must be taken to ensure the slot being exactly at right angles to the line of stroke.

A conveniently arranged gear of this type is shown in Fig. 54, applied to an engine with closed-in guides. A long stud B screwed into the crosshead pin carries a plate at D, and in this the vertical slot is formed with which engages the pin fixed in the end of the pendulum lever C. The length of the latter can be adjusted at F. The short lever H is flattened for a portion of its length at K, and on this are engraved several lines at right angles to the axis of the lever, and respectively in line with the corresponding series of holes for the connection of the cord. In adjusting the position of the lever H upon the cross shaft I, the hole is selected which will give the desired amount of motion to the indicator drum and the cord attached. The crosshead is placed in the middle of its stroke, and the arm H turned upon the spindle until the direction of the cord agrees with, or is parallel to, the line on K, in which position it is secured by means of the set-screw J.

Another arrangement which is sometimes found of service consists in bolting a flat strip to the crosshead, having a slot formed at its lower end with which the pin in the pendulum lever engages. The slotted bar is conveniently made of such a length as will allow the cord to be led off parallel to the line of stroke and direct to the indicator. Occasionally it may be found necessary to give a "set-off" to both the slotted bar and the pendulum lever, in order to clear the engine framing. It will be obvious that the gear shown in Fig. 45 may be modified in this way, and also that the brumbo pulley may be applied if desired. It is to be noted that by this method not only is the use of guide pulleys obviated, but also only a very short pendulum support is required, this adding greatly to the rigidity of the arrangement. Gears of this kind are specially suitable for engines of the girder-frame type, and also for vertical engines.

A convenient and accurate reducing gear is shown in Fig. 55 applied to the cylinder of an American locomotive. As will be seen, motion is transmitted from the pendulum lever nearly to the indicator by the sliding rod R. By this means only a very short length of cord is required—a distinct

advantage, especially for locomotive indicating. A pin fixed in the sliding rod engages with a slot in the upper end of

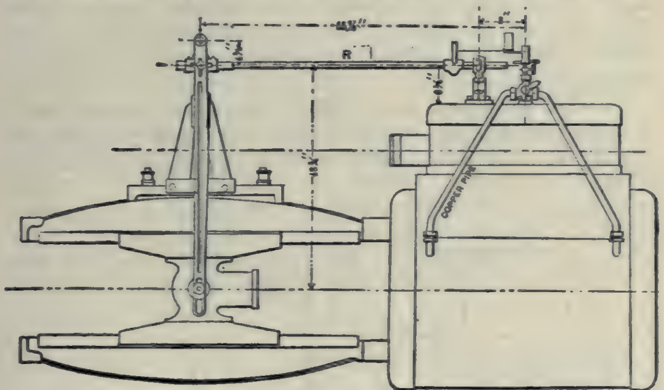

FIG. 55.

the reducing lever, while the connection with the crosshead pin is made as in Fig. 45. A slightly modified form of

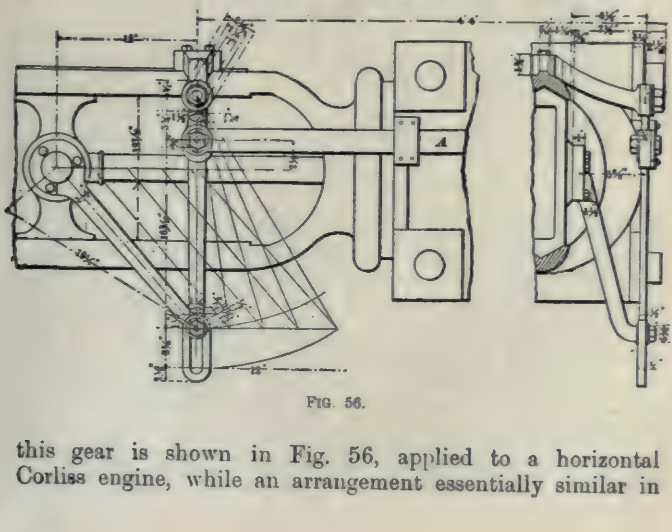

FIG. 56.

this gear is shown in Fig. 56, applied to a horizontal Corliss engine, while an arrangement essentially similar in

principle is shown in Fig. 57, applied to a vertical engine. In the latter case the connection with the crosshead is made by a telescopic arrangement, which increases the durability of the gear. Such an arrangement would be suitable for such a permanent reducing gear as is required for marine engines.

Another arrangement of pendulum lever very largely used is that in which the lower end of the main lever E is coupled to the crosshead by means of a vibrating connecting link F,

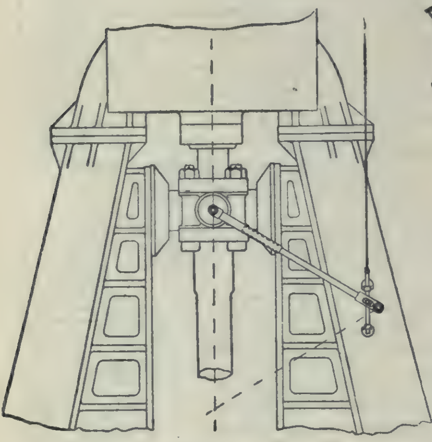

FIG. 57.

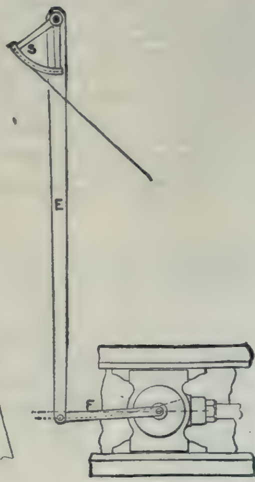

FIG. 58.

as shown in Fig. 58. This construction avoids the use of slots in levers, etc., which are frequently a source of trouble after some use. As in almost all lever reducing gears, the motion given to the indicator barrel is in this instance not an exact reproduction of that of the crosshead. This is for the most part due to the angular vibration of the connecting link. To reduce this to a minimum the link should be arranged to vibrate equally above and below the path of the crosshead pin, or, in other words, the path of the crosshead pin should bisect the versed sine of the arc described by the end of the pendulum lever. In Fig. 59, *e* and *f* are the

extreme and A the mean positions of the pin in the end of the lever. Joining *e f* gives *a* A as the versed sine of the arc *e* A *f*, and through the middle point *b* of this the horizontal path of the pin in the crosshead may be drawn. The pendulum lever must be vertical when the crosshead is in the centre of its path, and as it is evident that the error due to the swing of the connecting link will be less as the length is increased, it is advisable to use as long a link as convenient—preferably not less than one-half the length of stroke if at all possible. Frequently it will be found more convenient to use a bar depending from the crosshead (as previously suggested in connection with Fig. 45), to which the link can be attached. The brumbo segment is very often used with this form of pendulum lever, but, as will be

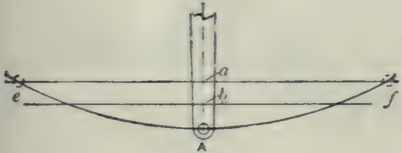

FIG. 59.

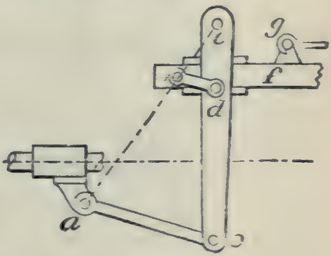

FIG. 60.

shown later, the reduction is then less accurately effected than with the pin connection.

A neat modification of this gear is shown in Fig. 60, wherein *f* is a rod moving in a slide parallel to the line of stroke of the engine, and carrying a pin *g*, to which the indicator cord is attached. The pendulum lever is connected with the slide by means of a short link *b d*, the length of which is in the same ratio to *a e* that *c d* is to *c e*. It follows that *c*, *b* and *a* always lie in the same straight line, and that the movement of *b* is an exact reproduction of that of the crosshead, but reduced in the proportion *c e* : *c d* = piston stroke : length of diagram. With this arrangement the connecting link *a e* can be placed at any convenient angle with the pendulum lever, as the restriction to be observed

in connection with the gear shown in Fig. 58 does not apply.

In the gear shown in Fig. 61 the lower end of the pendulum lever turns about a pin fixed in the crosshead, while another pin C, fixed in the upper end of the lever, moves up and down in the fixed slot guide, which must be in an exactly vertical position. The path of the pin P is shown in broken lines, and from this it is obvious that the rise and fall of the cord will be considerable, being nearly equal to the versed sine A′ S′ of the arc A S′ T. To reduce this error as far as possible, the cord should be arranged so that it is horizontal when the pin is midway in its vertical travel. A somewhat similar arrangement is shown in Fig. 62, in which the vertical slot-way is replaced by a vibrating

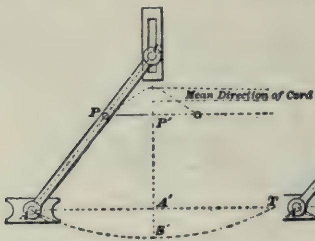

FIG. 61.

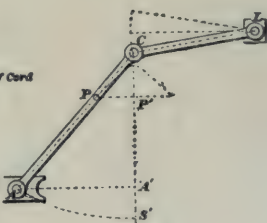

FIG. 62.

lever C L. If possible, the length of the latter should not be less than one-half the length of the pendulum lever C A, whilst the fixed centre L should be so placed that C L is as much below the horizontal at the extremities of the stroke as it is above it at mid-stroke. For convenience, the lever C L is sometimes placed on the other side of the vertical, but this should be avoided if possible. It will be noted that the brumbo pulley cannot be used with either of the gears last described.

Various other arrangements of pendulum lever reducing gears are to be met with, but a description of these would serve no good purpose, as they consist mainly of more or less fanciful modifications of the gears already described.

Pantagraph Reducing Gears.—Among the most accurate varieties of reducing gear are the modified forms of the

pantagraph, which are often used on long-stroke engines, and for which they are more suitable than any form of pendulum lever. Pantagraph reducing gears for temporary use are usually made of well-seasoned wood, but they do not give satisfactory results unless they are well made and carefully used. The joints should be large and provided with means for taking up the effect of wear, and when in use the lubrication of the several joints should receive careful attention. Fig. 63 represents a form of pantagraph motion-reducing device, sometimes styled the "lazy-tongs,"

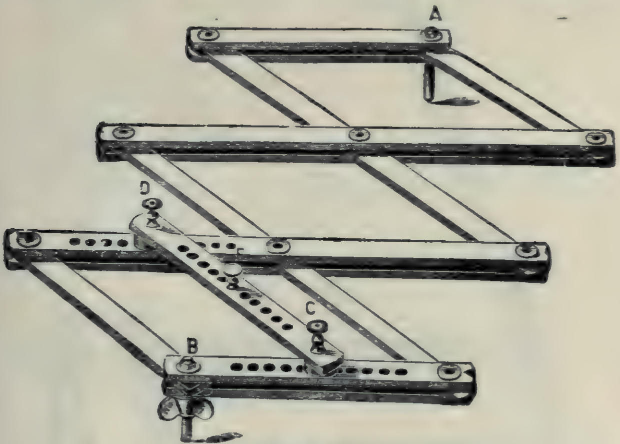

FIG. 63.

which has the merit of possessing a very wide range of application. It is therefore suitable for a travelling outfit, more especially since it has also the advantage of folding up into a very compact form. It consists of several wood strips of about 16in. long, 1in. wide, and $\frac{1}{4}$ or $\frac{5}{16}$in. in thickness, these being jointed together as shown. The pin A receives motion from the crosshead, the latter carrying a socket, or a drilled plate, in which the pin can turn freely. The fulcrum pin B is supported in any convenient manner by any available portion of the engine framing, or by a post or other support, which, for convenience, should be placed

about opposite to the mid-position of the crosshead. The cord, which must be led away parallel to the line of stroke, is attached to the pin E; and, to give various degrees of reduction, the position of the bar carrying this pin can be changed by inserting the screws C and D in the other sets of holes provided. In this way the ratio of B E to B A can be varied as desired, but it is of the greatest importance to note that the position of the pin E in the crossbar must

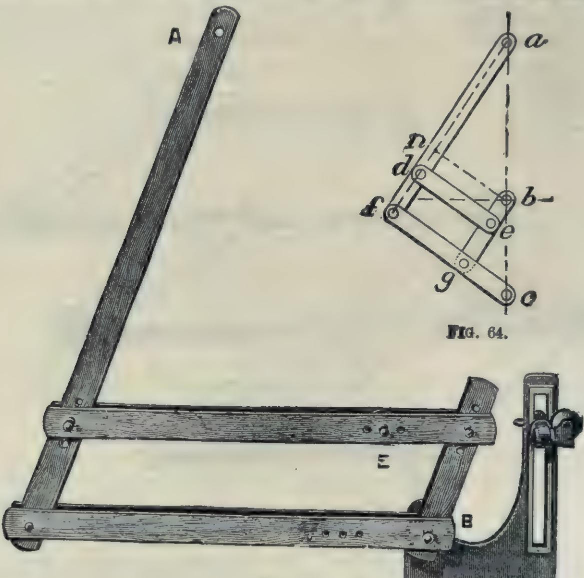

FIG. 64.

FIG. 65.

also be varied in order to keep B, E and A in one straight line, this latter being an indispensable condition with all forms of pantagraph reducing gear. The lazy-tongs may be arranged so as to move in a vertical, a horizontal, or any intermediate plane of position. It is decidedly unsuitable for high piston speeds, and although its portability and adaptability are valuable features of the arrangement, it must be admitted that it is a somewhat clumsy piece of apparatus, and one moreover which is very liable to get out of order.

The much more convenient pantagraph reducing gear shown in Fig. 64 is constructed of flat strips of iron carefully riveted together so as to allow perfect freedom of movement without lost motion. It is necessary that $d\ e = f\ g$, $d\ f = e\ g$, and that a, b and c lie in a straight line. The attachment to the crosshead is at a; c is the fixed fulcrum, the cord being led from b and in a direction parallel to the

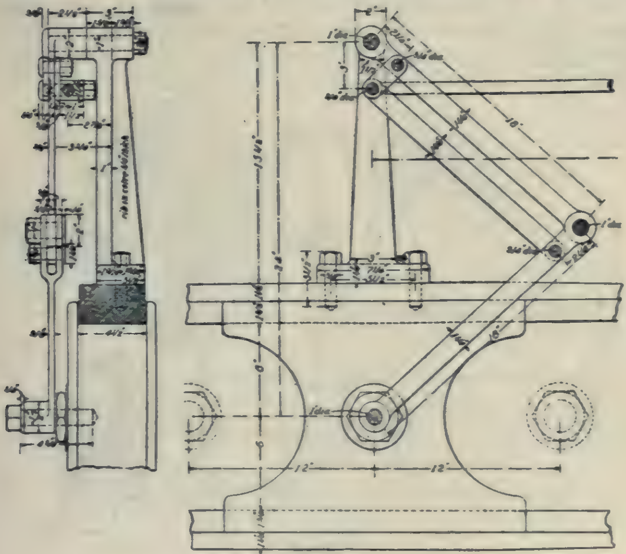

FIG. 66.

line of stroke. The crosshead movement is reduced in the ratio of $a\ c : b\ c$.

The adjustable pantagraph shown in Fig. 65 is made of well-seasoned strips of wood, pin-jointed together. The attachment to the crosshead is made at A, the cord being led horizontally from the cord pin E to the guide pulley shown. As before, A, E and B must be in a straight line.

A practical example of a more substantial pantagraph reducing gear, as applied to a locomotive, is given in Fig. 66. In this case motion is transmitted nearly to the

indicator by means of a round rod of steel $\frac{3}{4}$in. in diameter, the outer end of which is carried by a light bracket fixed to the engine cylinder. Another arrangement, shown in Fig. 67, is formed by interchanging the fulcrum and the point giving motion to the cord in the last example.

Reducing Wheels.—The reducing wheel is a very convenient form of gear, giving accurate results when correctly applied. These gears are usually arranged so that they may either be attached directly to the indicator, or fixed independently, as shown in Fig. 68. In either case a cord from the crosshead is led, *in a direction parallel to the line*

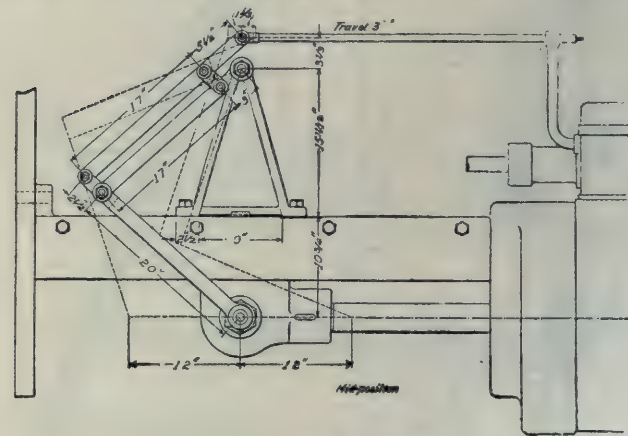

FIG. 67.

of stroke, on to the larger diameter of the reducing wheel. The cord from the smaller diameter is led directly to the indicator barrel, and when the gear is independently fixed, may run off in any radial direction, as may be convenient. It will be seen that the reduction of motion obtained by this device depends entirely upon the relative diameters of the two parts of the wheel. Therefore, in order to render the apparatus of general application, a number of pulleys or bushings is necessary, and a set of these is usually furnished which will meet all ordinary requirements. The cord leading to the crosshead is kept at a suitable tension by

means of a flat spiral or clock spring attached to the body of the wheel, and this can be regulated as desired, it being necessary to increase the tension on the cord with increasing piston speeds. Occasionally, with low speeds, it may be possible to dispense with the wheel-spring entirely, relying upon the drum spring of the indicator to maintain the requisite degree of tension on the cords. But this plan is not to be recommended.

It is obviously desirable, especially with high-speed engines, that the weight of the reducing wheel should be

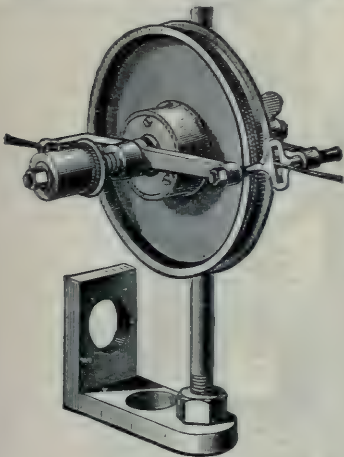

FIG. 68.

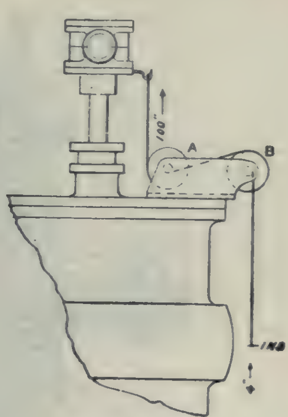

FIG. 69.

as small as possible, in order to diminish the effect of momentum, and to this end wheels of aluminium are now made, the weight of which is very small.

With long-stroke engines the difference in the diameters of the two portions of the reducing wheel may be such as to render a direct reduction inconvenient. A double reducing wheel may then be used attached directly to the indicator base. When mounted separately, such a gear will be found convenient for long-stroke engines generally, and, modified as in Fig. 69, it forms a suitable gear for oscillating engines, the example showing a

reduction of motion of from 100in. to 4in. The wheel A may be allowed to move laterally upon its stud in order to ensure even coiling of the cord. In another arrangement frequently adopted, studs are fixed at the ends of the guides, carrying pulleys grooved for $\frac{1}{4}$ or $\frac{3}{8}$in. cord. The centres of the pulleys are in a plane parallel to the line

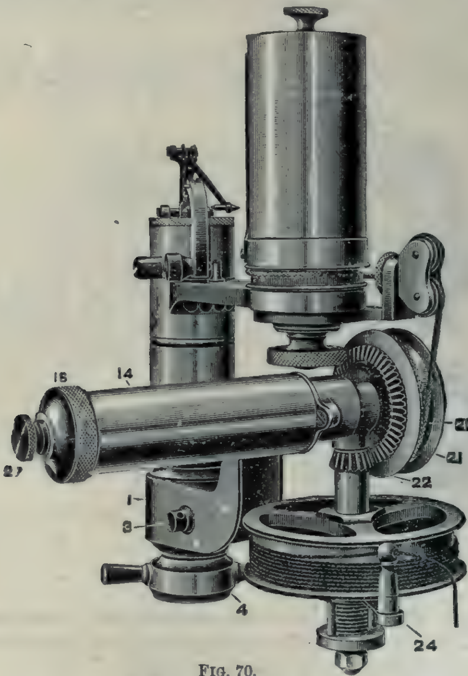

FIG. 70.

of stroke, and the cord which is stretched tightly over them is joined by means of a short iron plate, having holes at each end for the attachment of the cord, and a large hole in the centre which fits on a pin projecting from the crosshead. The boss of the pulley nearest the cylinder is lengthened so that the indicator cord may coil upon it, and in this way the reciprocation of the crosshead gives the

required motion to the drum of the indicator. The cord may be led off directly to the instrument without the intervention of guide pulleys, but care must be taken to see that the cord coils evenly upon the boss of the reducing wheel.

The Crosby reducing wheel, shown in Fig. 70, is attached directly to the indicator cock, being fixed in any desired position by the screw 3. The return movement is effected by a helical spring in the case 14, the tension being adjusted by releasing the screw 27 and turning the milled head 16. The cord is coiled on the pulley 20, at each side of which are discs 22 and 21. On removing the latter, it will be found that the pulley is made up of a number of concentric rings each of which is marked for a certain stroke with a range of from 14 to 72in. A suitable pulley having been selected, all outside it are removed and the retaining disc replaced. A coiling gear is fitted below the large wheel. The cord guide 24 is adjustable so that the cord from the crosshead (or guide pulleys) may lead on to the wheel without rubbing in the hole. All the moving parts are as light as possible, the stroke pulleys being of aluminium, while ball bearings are employed to ensure easy running. On a 2in. drum the diagram will be about 4in. long, and on a 1½in. drum, about 3in. long. When fitted with a detent, the arrangement is specially useful for indicating gas engines.

A very compact reducing arrangement known as the "Houghtaling" reducing gear is shown in Fig. 71 applied to the Tabor indicator. As will be seen, a quick-threaded worm carried by a bracket attached to the base of the instrument, gears with a worm wheel on the lower part of the paper drum. The pulley upon which the driving cord is coiled, runs loose on the worm spindle until a clutch is thrown into gear, when both worm and pulley move as one. An adjustable spring ensures the prompt return movement of the pulley; the drum spring also assists this action, as the pitch of the worm wheel is such that it will drive the worm in either direction. Cord pulleys of various sizes are supplied to suit lengths of strokes up to 6ft., and in using the motion, a pulley is selected the circumference of which is from one-fourth to one-fifth the length of stroke.

The clutch collar and clutch are then added, the cord wound upon the pulley a sufficient number of times to allow the engine to make its stroke without completely unwinding it, and the loose end of the cord attached to the crosshead with the engine on the back centre. The clutch is so arranged that it cannot catch on the backward stroke, but will do so within a half revolution of the pulley on the forward stroke.

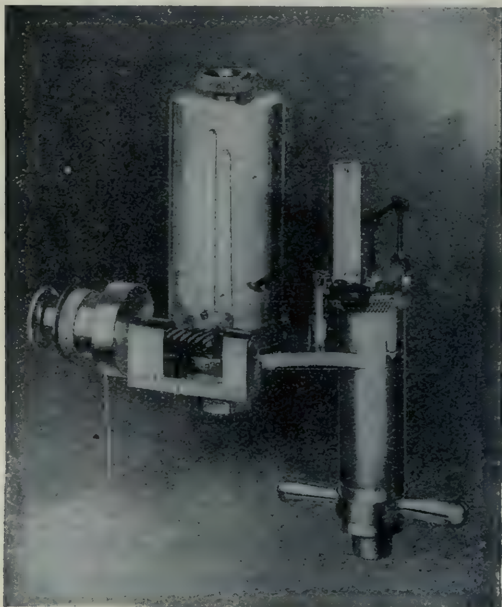

FIG. 71.

In order that the drum when in motion may clear the stop on the return stroke, it should be turned forward $\frac{1}{4}$ or $\frac{1}{2}$in., and if the clutch is then thrown in, the drum will start off on the next forward stroke. The milled nut at the top of the drum enables the latter to be turned at starting, and after the diagram has been taken, the clutch may be disengaged by holding this nut. The cord from the crosshead must of course be first led in a direction *parallel to the line of stroke,*

but this may often be accomplished without using guide pulleys, by fixing a bracket or standard to the crosshead, and attaching the cord to this so that it may be led directly to the pulley.

Various methods of attaching the cord to the crosshead are indicated in Fig. 72.*

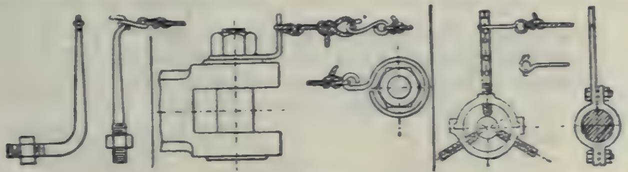

FIG. 72.

Gas Engine Reducing Gear.—The absence of a crosshead in the gas and oil engine, as usually constructed, prevents the direct application of many forms of steam engine reducing gears. In some cases it is possible to fix a bar to the piston, the pendulum lever being connected to a pin in the outer end of this bar as indicated in Fig. 51, which shows

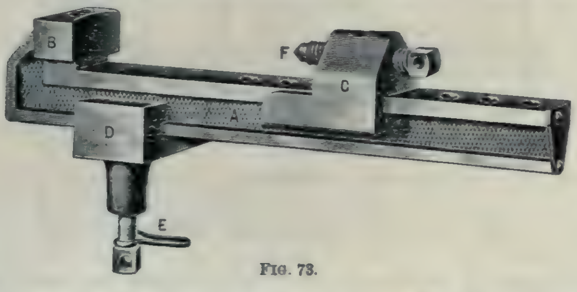

FIG. 73.

one form of gas engine reducing gear. The same means may be necessary to provide an attachment for the cord of a reducing wheel. Frequently, however, an attachment to the crank shaft provides the most satisfactory arrangement. Such an attachment, known as the "Simplex" indicating gear, is shown in Fig. 73. It consists of a beam A of

* From Haeder's "Der Indicator."

aluminium having a hollowed lug B and an adjustable carriage C. On the other side of the beam is an adjustable crankpin D, carrying a hook E to which the cord is connected. The engine is placed on the full outstroke dead centre, the beam clamped on the end of the crankshaft by the pinching screw F so that it lies parallel with the centre line of the engine, and the crankpin D adjusted to give the desired length of diagram. The cord must, of course, be led in a direction parallel to the line of piston stroke.

A reducing gear with a wide range of adaptability and

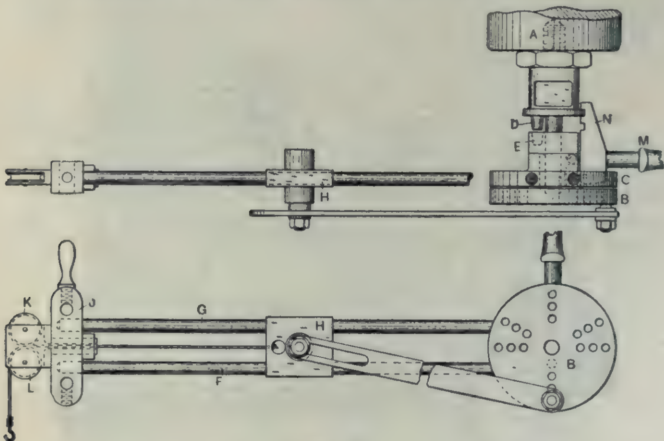

FIG. 74.

particularly useful for gas engine work is the Graham "Nonstop" gear made by Messrs. Schäffer and Budenberg and shown in Fig. 74. It is applied to the engine crankshaft, the latter being drilled and tapped to receive a stud at A. An extension of this stud forms a bearing for the crank disc B with its sleeve E, in the end of which is a hole to receive the pin D. On the outside of the sleeve E is a split flanged bush C held together by the screwed ends of two guide rods F, G, on which slides the crosshead H. The outer ends of the rods are secured in a yoke J which can be fixed in position by any convenient means, the cord from

the crosshead being led away through the adjustable guide pulleys K and L.

The crank disc has a number of holes at 90° and 120°, to suit the different crank angles of a multi-cylinder engine, these holes being spaced radially so as to give three lengths of diagrams :—$1\frac{3}{4}$in., $2\frac{1}{2}$in., and $3\frac{1}{4}$in. ; one connecting rod is provided for each length. It will be understood that the slot in the connecting rod enables the ratio of lengths of crank to connecting rod to be made the same as in the engine to be indicated. A spring-controlled handle M hinged in the bush C, has an extension N with two slots, each capable of engaging with a collar on the stud. To

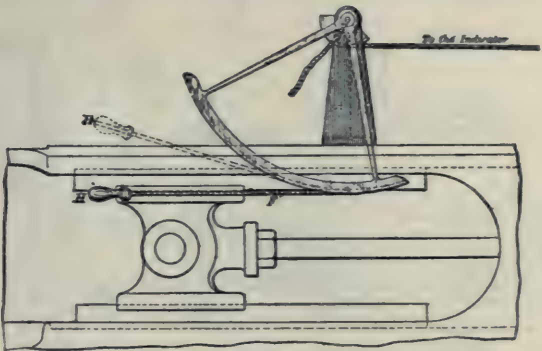

FIG. 75.

start the gear, N is disengaged from the collar by pressing the handle M ; then B and C together are moved inward until engagement with the pin D is effected, causing the disc B to revolve. The handle is then released, so that the lower slot in N now engages with the collar, retaining the gear in its running position.

Miscellaneous Devices.—In addition to the foregoing arrangements, numerous other reducing devices have been proposed from time to time, many of which are too complicated for practical use. It must be admitted, however, that several of these special devices afford very neat solutions of the problem under consideration, while they frequently suggest modifications of the more general forms

of reducing gear which may be necessary in order to meet the requirements of particularly difficult cases.

A modification of the reducing wheel, which may be termed a reducing segment, is shown in Fig. 75. The cord F may be conveniently disconnected from the claw fixed in the crosshead by raising the handle H when the return stroke is commencing. A spiral spring acting on the pin which supports the segment, effects the return movement.

Two modern reducing gears are shown in Figs. 76 and 77, the first being an arrangement used on a short-stroke

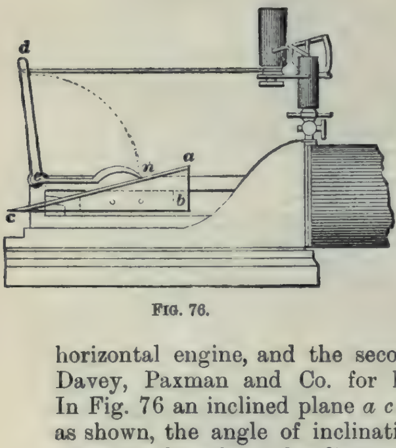

FIG. 76.

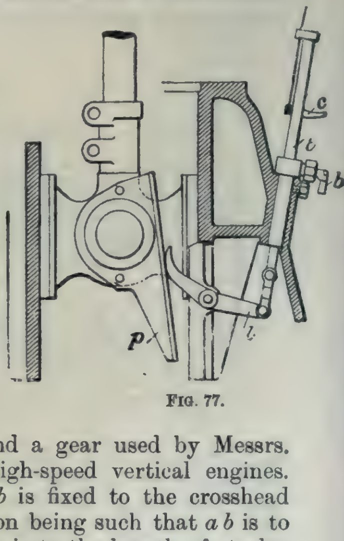

FIG. 77.

horizontal engine, and the second a gear used by Messrs. Davey, Paxman and Co. for high-speed vertical engines. In Fig. 76 an inclined plane *a c b* is fixed to the crosshead as shown, the angle of inclination being such that *a b* is to *c b* as the length of the diagram is to the length of stroke. The end *n* of an equal-armed bell-crank lever *d e n* (turning on a fixed pin *e*) rests upon the incline, while the indicator cord is attached at *d*. The arm *d e* should be vertical when the crosshead is in the centre of its travel. In Fig. 77 the lever *l* is nearly straight, and is coupled at its outer end to a sliding rod working in the tube *t*, the latter being slotted to admit the piece *c*, which forms the end of the sliding rod, and to which the indicator cord is attached. A spiral

spring (not shown) fixed in the upper part of the tube *t* keeps the claw of the lever *l* in contact with the inclined plate *p*. A spring catch *b* enables the gear to be readily thrown out of action.

With engines having the working parts enclosed, it is as a rule impossible to apply any of the gears previously described. Special arrangements are therefore necessary, and these often take the form of an eccentric mounted upon the crankshaft outside the engine casing. A convenient gear of this kind is shown in Fig. 78. In this case, the eccentric strap is coupled at its upper part to one end of a swinging lever B, the other end O of the latter turning upon a fixed

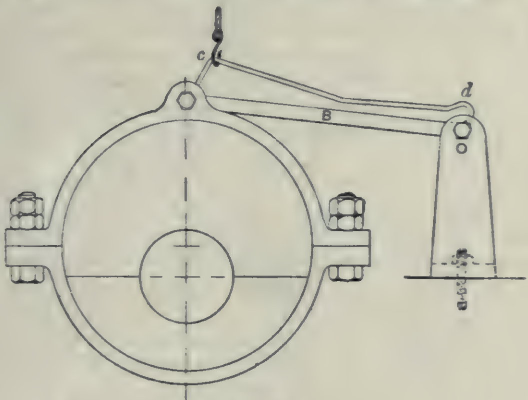

FIG. 78.

pin carried by a bracket bolted to the engine frame. The eccentric must be mounted upon the shaft so that the limits of the cord travel occur as the crank passes the dead centres. The cord is not directly connected to the lever B, but is hooked on to a bent wire attached thereto as shown. When the hook is at *c*, full motion is given to the indicator drum; but when the hook is drawn back to *d*, the cord receives no movement. In other arrangements, the lever B is prolonged, and the cord attached to the free end, thus reducing the travel of the eccentric.

When the crankshaft is available for the purpose, motion

is sometimes given to the indicator cord by a pin fixed into the end of the shaft, this being placed sufficiently out of the centre to give the required amount of travel. With single-crank engines fitted with a centrifugal oiling tube, the same end may be attained by attaching a clip to the latter carrying a small pin. Special care must be taken to so place the driving pin that the extreme points of the travel of the cord are reached precisely when the engine is passing the dead centres. On consideration, it will be seen that by reason of the angular vibration of the connecting rod, the motion given to the drum will deviate from that of the piston to an extent which will be greater as the ratio of the lengths of the connecting rod and crank is less. This error may be entirely eliminated by coupling to the pin one end of a link, the other end (to which the cord is attached) being constrained to move in the plane of the shaft centre as in Fig. 74. For correct results, the throw of the pin and the length of the link must be proportional to the lengths of the crank and connecting rod.

In *beam engines*, one of the radius rods will frequently form a convenient reducing gear, the cord being attached at such a point as will give the required travel, and being led *vertically* downwards either to the instrument or to a guide pulley if one is needed. A less correct plan is to connect the cord to a pin fixed in the beam and sufficiently near the gudgeon to give the required amount of movement. In this case, when the beam is horizontal, the line joining the centres of the gudgeon and the pin must be at right angles to the direction in which the cord is led off. Occasionally the diameter of the gudgeon may be such that the required movement can be given to the cord by coiling one end of it around the gudgeon; in this case, the cord may be led off in any convenient direction. In the marine type of vertical engine, the cord may often be very conveniently attached to the air-pump lever. Occasionally it is possible to adopt the same plan in horizontal engines when the air pump is driven from the crosshead by mean of an L-lever.

Reducing Gear for Valve Chest Diagrams, etc.—When diagrams are taken from the steam pipe or valve chest, the high-pressure cylinder reducing gear should be employed. For diagrams from the receiver either the high or low-

pressure cylinder gear may be used; while for diagrams from the condenser or air pump (when this is driven from the low-pressure crosshead), the same reducing gear should be used as for the low-pressure cylinder.

Cords and Wires.—In arranging any form of reducing motion, it is very desirable to avoid the use of long reaches of cord. Firstly, for the obvious reason that the longer the cord the greater the stretch and consequent distortion of the diagram; and secondly, because long cords sag and vibrate considerably, especially at high speeds. Strips of wood may often be employed to transmit the motion to a point quite close to the instrument. Steel, copper, or annealed iron wire is often substituted for cord with advantage, a piece of cord being inserted where a guide pulley has to be passed. Iron and steel wire, however, has the disadvantage of being liable to "kink," and is generally not so convenient to manipulate as cord. The wire used must not only be free from kinks, but it must also be quite straight, as any bends in it may lead to greater errors than would be the case with cord. Tests have shown that steel wire No. 28 B.W.G. (0·014in. in diameter) yields about 0·0012in. per foot; while No. 36 B.W.G. (0·004in. in diameter) yields about 0·003in. per foot for each pound of tension. The balance of convenience is in favour of the use of cord, and if the precaution is taken to thoroughly stretch it before using, the error will be inconsiderable. By hanging a heavy weight from one end of a length of cord for a day or so, the remaining elasticity is usually insufficient to allow the cord to stretch more than from 0·007 to 0·009in. per foot—according to the diameter—for each pound of tension. The special kind of closely-braided cord supplied by the various instrument makers varies from about $\frac{1}{12}$ to $\frac{1}{16}$in. in diameter, and if used without any preliminary stretching will be found to yield considerably more than the above amounts. On the other hand, cord which has been used for some time appears to take a "permanent set," this reducing the yield per foot to about 0·006in. per pound of tension. In any case, the application of a little beeswax, well rubbed in by means of a notched piece of wood, will be found beneficial, as it renders the cord less susceptible to the influence of water or steam.

Composite cords are also used in which wire and fibrous material are used in conjunction, with a view to securing freedom from stretch combined with flexibility. Messrs. Dobbie McInnes, Limited, supply a good cord of this kind, composed of six strands of No. 36 B.W.G. steel wire running straight through a double flax tube which is woven over them.

When the use of long stretches of cord or wire is unavoidable, it is preferable to lead it past the indicator, connecting it to a spiral spring or rubber band attached to the engine framing or other rigid support. In this way the cord transmitting the motion to the paper drum is maintained in constant tension, while the work of the drum spring of the indicator is reduced to the minimum. A short length of cord attached to the drum is provided with a light wire hook which, even at very high speeds, may be readily slipped into or out of a loop tied in the driving cord. When wire is used instead of cord, a small button is fixed upon the wire, and as the hook attached to the short drum cord is forked, the connection is effected very readily.

CHAPTER VIII.

ERRORS OF INDICATOR REDUCING GEAR.

MENTION has already been made of the fact that very many of the indicator reducing gears in general use give results which are far from accurate. It is therefore desirable to briefly consider the nature and extent of these errors, giving more particular attention to those of the pendulum lever class so extensively employed, by making a graphic comparison of the movements of the engine crosshead and the paper drum of the indicator.

In Fig. 79 the various forms of pendulum gears are shown diagrammatically, being lettered from A to G, while Fig. 80 is a series of diagrams giving a graphic comparison of the movements (on a reduced scale) of the crosshead and the paper drum of the indicator, for successive tenths of the stroke in each case. In obtaining these diagrams, the author in the first four examples has taken the effective length of the pendulum rod at the centre of the stroke as 36in., the stroke 24in., while the reduction of movement is such as to give a diagram approximately 4in. in length. These proportions are not in any way unreasonable, and in order to give every advantage to the gears, the pendulum is understood to be exactly vertical when the crosshead is at the centre of its travel, thus reducing the error as much as possible. The exceedingly small error due to the vertical movement of the pin to which the cord is attached in A, C, and E is disregarded, as in all ordinary cases it is quite inappreciable. The dimensions selected for E, F, and G are those used by a firm of high-class engine builders, and are:—Length of lever, 24in.; stroke, 24in.; connecting link, 8in. · diagram,

4in. The gear shown at A and B probably owes much of its popularity to its simplicity, but from the corresponding comparative diagrams A′ and B′, it will be

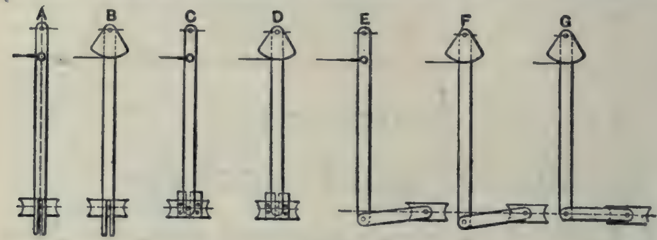

FIG. 79.

seen that the results obtained are by no means satisfactory, although the error is not so marked in B, where the

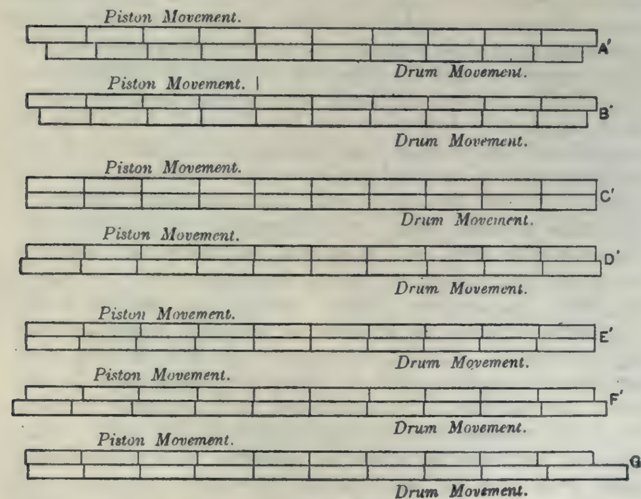

FIG. 80.

brumbo pulley is employed for the cord connection. It will be noted that the error rapidly increases as the ends of the stroke are approached, pointing to the desirability of

having the pendulum lever as long as possible whenever this form of gear is employed. In the modification of this gear, shown in Figs. 48 and 49 (page 89), the errors are precisely similar in character to those incurred with A, but as a rule they are more pronounced, as the effective length of the pendulum is generally small, owing to the standard carrying the upper support being kept as short as possible.

The gear shown in C and D is not very generally used, but an inspection of diagrams C′ and D′ shows that it forms a much better reducing gear than either A or B. Indeed, when the cord is attached to a pin as in C, the reduction is perfect, and even with the brumbo the error is not so great as in the first cases selected. It is evident that this type of gear can be substituted with advantage wherever A or B is already in use, and if C is used the pendulum may be considerably shorter than usual without introducing any practical error whatever.

The link-connected gear shown in E, F, and G is almost invariably used with the brumbo cord connection, but the comparison afforded by diagrams E′ and F′ shows this to be a mistake, the "pin" cord connection E involving an error less in amount and more uniformly distributed than when the brumbo is used.

The desirability of arranging matters so that the connecting link vibrates equally above and below the horizontal, has been already emphasised, but frequently the construction is such that when the crosshead is in the centre of its path the pendulum is vertical and the connecting link horizontal, as shown at G. The effect of this, using the brumbo cord connection, is exhibited in diagram G′, while the distortion produced in an ordinary diagram by the last arrangement is shown in Fig. 81. As will be seen, the displacement of the admission end of the diagram is such as to introduce a considerable amount of error in the calculation of the mean effective pressure, as the area of the figure is materially increased, and at the same time the general character of the diagram is shown to differ largely from that which should be obtained.

Of the other pendulum gears previously described and not dealt with in the foregoing remarks, Fig. 50 (page 89)

is exceptionally inaccurate. This will be evident when it is considered that as mid-stroke is approached the virtual length of A P increases, while that of P C decreases, to which has to be added the error due to the extensive vertical movement of the cord pin. The gears shown in Figs. 55, 57, and 60 give perfectly accurate reductions of motion. With the gear shown in Fig. 61, the vertical movement given to the cord pin introduces an error which will become less as the length of the lever is increased ; but with a lever of reasonable length, and a fair length of cord between it and the guide pulley, the error introduced is not great. When the upper end of the lever is hung from a fairly long rocking lever, as in Fig. 62, the angular movement of the latter partly neutralises the error due to the rise and fall of

FIG. 81.

the cord pin. If, however, the rocking lever is placed on the opposite side of the centre, the error will be measured by the sum, instead of the difference, of these angular movements, in which case, especially if a short cord is used, the reduction will be very inaccurately effected.

With pantagraph reducing motions properly applied, no errors are theoretically possible. In practice, however, and especially in the case of the lazy-tong arrangement (Fig. 63, page 97), wear of the joint pins in the wood strips causes lost motion or "play," which may introduce serious errors. Care must be taken to correctly locate the cord pin (see page 87), and to lead the cord away in a direction *parallel to the line of stroke both vertically and horizontally.* Neglect of this latter precaution is a frequent source of

error. Stretching of the cord and the tendency of the latter to coil unevenly on the pulleys, and by "mounting" to distort the diagram, are also objections. The latter defect has been removed in later forms of reducing wheels by adding a cord guide which ensures even and regular coiling; while, as regards the first-named objection, it is to be noted that as the movement of the main cord is greater, the stress on it will be proportionately less than on cords which drive the drum directly.

The gear shown in Fig. 76 is convenient, as it can be very readily thrown out of action, but the reduction will not be correctly effected if motion is given to the lever by an angle piece as shown. The error, it may be noted, could be reduced by lowering the fulcrum e so that $e\ n$ is parallel with the inclined surface $c\ a$ when at mid-stroke. As shown, however, the acting surface $c\ a$ requires to be slightly curved in order to give equal angular movements to the arm $e\ d$ for equal horizontal movements of the cross-head. In place of curving the guide piece, the toe of the lever may be made of such a shape as will give the desired motion (Fig. 77).

The precautions necessary to be observed in eccentric gears, direct driving from the crankshaft, etc., have been noted in describing these.

From the foregoing considerations, it will be apparent that slotted pendulum gears generally give more or less inaccurate reductions of motion, while they are unsatisfactory from a mechanical point of view. The distortion of the diagram, especially with gas and oil engines, or with steam engines when the cut-off occurs early in the stroke, is frequently considerable, except in the case of the gear shown in Fig. 52 (page 91). Levers driving cord-motion sliding bars by means of slots and pins are to be avoided, as any lost motion in such gears tells directly upon the motion of the paper drum, as will also any vibration of the standards or other supports from which they are suspended. Pantagraph gears and reducing wheels are generally convenient, as they can be readily adapted to suit varied requirements. Finally, for a permanent form of reducing gear, a selection may be made from the arrangements shown in Figs. 54, 57, 60, and 78.

CHAPTER IX.

THE USE AND CARE OF THE INDICATOR.

THE necessary arrangements for the attachment of the instrument and for giving motion to the paper drum having been completed, the manipulation of the indicator has now to be considered.

General Hints.—It is important to keep well in view the fact that the accuracy of the diagram is largely influenced by the care bestowed upon the instrument, and especially is it necessary to see that the cylinder and piston are quite clean and well lubricated. Attention should also be given to the various joints in the pencil motion, to ensure that while no play exists in the mechanism, the movement is perfectly free throughout the whole travel.

Assuming the stopcock to be screwed into the cylinder or piping, as the case may be, steam should be allowed to flow through the connections so as to clear them thoroughly before placing the instrument in position.* This will remove any dirt, grease, scale or sand which may have collected, and which would otherwise find its way into the cylinder of the indicator. The presence of dirt in the cylinder causes the piston to move spasmodically, and will manifest itself in the diagram by giving a serrated or step-like appearance to the expansion line, the effect being similar to that produced by piston friction, already referred to (page 59). Water in the instrument will have a somewhat similar influence upon the diagram.

When the instrument is running, any want of smoothness of motion can be detected by placing the finger lightly upon

* In condensing engines this must obviously be done during the steam stroke of the engine.

the pencil lever near the back fulcrum. As this may, however, be due to friction of the joints of the pencil motion, this point should be decided by removing the instrument from the engine, taking it apart, and placing the piston and rod in the upper part of the cylinder, while the thumb covers the opening at the base of the cylinder through which the steam enters the indicator. Then with the latter held quite vertically, the piston should fall *very slowly* to the bottom of the cylinder as the air escapes past the piston. Unless it will do this, both cylinder and piston should be thoroughly cleaned, and for this purpose a piece of fine sponge, or even tissue paper, will be found preferable to cotton waste, as the fibres of the latter often cause trouble. Should the surfaces of the piston and cylinder have become scored, they should be freely lubricated and a spring sufficiently weak to allow the piston its full travel should be inserted. Then with the pencil movement disconnected, the instrument should be allowed to run under steam for a short time. This will generally remedy any slight scoring and secure the desired freedom of movement. But no putty powder or other grinding material should be used. Occasionally, by slightly unscrewing the cap and slowly tightening it again, the piston will be found to work more freely.

Having ensured the satisfactory working of the piston, the pencil movement should next be examined. If held vertically, the pencil lever, when raised and released, should drop with perfect freedom to its lowest position. If there is any tendency to catch in any part of its path, the several joints should be examined and the levers looked over to ensure that none are "sprung" or distorted in any way. The instrument may then be put together, but without the spring, the lever raised, and the opening at the base of the cylinder closed by the thumb as before. With the instrument held vertically, the whole of the moving parts should descend slowly, but more rapidly than in the previous case. On the removal of the thumb while the piston is descending, the whole should fall instantly to the lowest point in the travel.

In order to test for play or lost motion, a fairly strong spring should be placed in the instrument, when

any unrestrained movement can be detected by carefully feeling at the outer end of the pencil lever. To locate this error, the several joints of the pencil movement, the connections of the piston, piston rod, and spring, as well as the fit of the swivelling collar carrying the pencil movement, should be each severally examined.

If the instrument has not been used for some time, the joints of the pencil movement may work stiffly owing to the gumming of the oil with which they have been lubricated. Soaking the pencil movement in petroleum or naphtha will generally be found the most expeditious method of restoring the desired freedom of motion, and only with old instruments, or those which have been used carelessly, will more drastic measures be needed. Generally, however, the small joint pins should not be tampered with, and any serious defect either in this or in any other part of the instrument should be left for the makers to remedy. A slight amount of side play is generally allowed in these joints to ensure freedom, and if they are oiled occasionally with watch oil—a small bottle of which usually accompanies the instrument—they will not ordinarily give any trouble.

Occasionally the pencil, when moved vertically upward by lifting the piston, may be found to bear unevenly upon the paper, making a thicker line at the upper than at the lower part of its travels, or *vice versâ*. Assuming the pencil movement to be correct, this will probably be due to the drum spindle being out of parallel with the indicator cylinder. To test this, the spring should be removed, and the pencil, finely pointed, should be adjusted so as to just allow light to be seen between it and the paper. Then by raising the piston slowly throughout its whole range of movement, it can be seen whether the space between the pencil and paper remains the same, or, if not, in what way and to what extent it varies. It is next advisable to ascertain whether the barrel is distorted or improperly centred. For this purpose, the pencil should be kept at one height, and the barrel slowly rotated by pulling the cord, while the space between the pencil and paper is carefully observed. By repeating this test with the pencil at various heights, any distortion of the pencil barrel can be detected.

Another test which it is sometimes advisable to make consists in drawing a fine vertical line by moving the piston upward, while the paper drum is held stationary at about mid-travel. Then allowing the piston to fall to its lowest position, a horizontal line is drawn by turning the drum while the pencil is at rest, care being taken to guard against any slight lifting of the drum occurring owing to vertical play upon the spindle. Removing the paper from the drum, the two lines so drawn are then to be tested for perpendicularity. Fig. 82 shows a convenient method of doing this, which consists in setting-off from b equal distances $b\,a$ and $b\,c$, and then from a, with any convenient radius, cutting $b\,d$ in d, and from c, with the same radius, cutting $b\,d$ again. Unless these two cutting arcs intersect each other and the line $b\,d$ in the

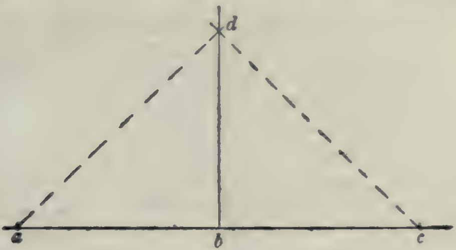

FIG. 82.

same point, either the pencil movement is incorrect or the drum spindle is bent. This test may be extended by drawing a *series* of horizontal lines as above, and testing them in the manner described.

The spindle upon which the drum turns must be lubricated from time to time, special attention in this respect being needed for high-speed work. It should be remembered that this part of the instrument is subjected to considerable wear, and being in some degree out of sight, it is liable to be neglected. The pin of the guide pulley also requires occasional attention.

Lubrication of the Piston.—Considerable diversity of opinion exists in regard to the lubrication of the indicator piston, and the character of the lubricant which should be employed. Some operators maintain that sufficient

lubrication is supplied by the steam, but the majority will be found to agree with the author that a good cylinder oil applied to the surfaces in contact gives the most satisfactory results. A light, thin-bodied oil is sometimes advised, but more frequent applications will be required, and in general it is not so suitable for the purpose as porpoise, castor, or some similar variety of *good* oil. As to the frequency of application, much will depend upon the work in hand. For low-speed engines with steam of moderate pressure, ten or twelve diagrams may be taken before it is necessary to again lubricate the piston. But for higher speeds and pressures, oil should be applied twice as often, while in the case of gas and oil engines, still more frequent attention will be required in consequence of the very high temperatures dealt with.

Strength of Spring.—The strength of spring which it is desirable to employ will, under ordinary circumstances, be determined by the initial pressure to which it is likely to be subjected, and the speed of the engine. At low speeds, the only factor to be considered is the greatest pressure to be provided for, and in this case, reference to the table given on page 123 will enable a suitable spring to be selected. In this table, the first column gives the "number," "scale," or "strength" of the spring, which number will be understood to designate the amount of pressure per square inch necessary to compress the spring sufficiently to move the pencil one inch. Thus, a spring numbered 40 will require a pressure of 40lb. per square inch to cause the pencil to move one inch, or, conversely, $\frac{1}{40}$th of an inch in vertical measurement of the diagram will represent a pressure of 1lb. per square inch.

The pressures given in the several columns of the table are the maximum to which the springs should be subjected, but for ordinary working conditions many makers advise an allowance of 10 to 20 per cent. on the figures given. It should be noted that, in some instances, springs above a certain strength are not made to indicate pressures below the atmosphere. This is a matter which should not be overlooked, as, with these springs, if the pressure should fall materially below this limit, it will not be correctly indicated, owing to the piston "bottoming" in the cylinder

TABLE I.—STANDARD INDICATOR SPRINGS.

Scale of Spring (lb. per sq. in. for 1 in. pencil movement).	Maximum Pressure in Lb. per Sq. In. above Atmosphere.										
	Richards.	Thompson (S. & B.).			Tabor.	Darke.	Crosby.	McInnes-Dobbie.		Simplex.	Casartelli.
		Large	Small.	"1911."				Large.	Small.		
8	10	8	—	3	10	—	14	3	—	—	7
10	16	12	—	7	15	—	—	7	3	15	13
12	22	18	6	12	—	6	21	12	—	—	17
16	35	30	12	20	28	13	28	21	10	—	30
20	47	40	20	30	40	20	35	30	15	45	40
24	60	50	—	38	48	27	42	39	20	—	52
30	78	68	24	—	70	37	52	52	30	75	70
32	100	75	45	55	75	40	—	72	35	—	88
40	125	90	60	70	95	55	70	90	60	120	110
48	150	125	70	90	112	70	—	108	—	—	130
50	—	—	—	—	120	72	88	—	75	150	135
56	175	—	80	100	—	80	—	126	85	—	155
60	187	150	—	110	140	90	105	135	90	180	160
64	200	—	100	120	152	95	—	144	100	—	175
70	—	—	—	—	—	—	—	—	105	210	—
72	225	180	—	140	—	110	—	162	—	—	195
80	—	200	125	155	180	125	140	180	120	240	210
100	—	250	150	200	200	160	175	225	150	300	250
120	—	—	180	240	240	195	210	—	180	—	300
150	—	—	—	300	290	—	260	—	225	—	—
180	—	—	—	360	—	—	315	—	—	—	—
200	—	—	—	—	375	—	350	—	300	—	—

Other springs supplied, with their maximum pressures, are as follows:—Small Thomson, 132 (200lb.); 160 (240lb.). Darke, 88 (140lb.); 96 (150lb.); 115 (186lb.); 128 (209lb.). Crosby, 4 (7lb.); 250 (435lb.); 300 (525lb.). Small McInnes-Dobbie, 140 (210lb.); 250 (375lb.). Simplex, 15 (30lb.); 25 (60lb.); 55 (165lb.); 90 (270lb.).

In most cases the range of the spring is from 15lb. per square inch to the tabulated pressure. The principal exceptions are the springs for the Richards, Casartelli, Large McInnes, Small McInnes, and Simplex Indicators. In the first, second, and third springs from 32, in the fourth from 40, and in the fifth from 35 and upwards, are scaled for pressure only.

of the instrument. In such a case, the back-pressure line would appear perfectly straight, utterly destroying the value of the diagram as a measure of the power developed, or as indicating the action of the steam when the pressure falls below that of the atmosphere. Should it occur, however, that no one spring is available which will cover the required range above and below the atmosphere, such a diagram as described may first be taken, after which, by substituting a weaker spring, another figure may be obtained, which will correctly show the "vacuum" side of the diagram. By reducing the ordinates of the latter diagram to the same scale as the first, the two may be combined; but if only required as a measure of the horse-power, the areas above the atmospheric line in the first diagram, and below it in the second, may be separately computed and added together. If this method is pursued, however, care must be taken to protect the weak spring from over-compression, and for this purpose a suitable length of brass tube may be slipped over the piston rod.

In general, it is advisable to use the lightest spring which may be employed with safety, since the diagram will then be as high as possible. Exceptions to this rule occur when indicating engines in which the compression may exceed the initial pressure, as sometimes happens with engines fitted with automatic cut-off gear, and in locomotives well "notched up," when, unless the indicator is provided with a safety stop, limiting the compression of the spring, the latter may be injuriously affected. On the other hand, if too weak a spring is employed, the steam line of the diagram will be entirely misleading, as a horizontal line will be drawn, while the spring is forced against the stop, and thus neither the action of the steam nor the maximum pressure will be correctly recorded.

With high speeds, stronger springs should be used in order to counteract the disturbance produced in the diagram by the inertia of the pencil mechanism, piston, etc., since the energy of these parts when in rapid motion corresponds in effect to an increase in the initial pressure. Hence it is desirable for speeds of from 150 to 300 revolutions per minute with the Richards instrument, or from 200 to 400 with the more modern indicators, to multiply the values

given in the table by 0·8, and for speeds of from 400 to 600 revolutions by 0·6, in order to obtain the maximum pressures to which it is desirable the springs should be subjected. Much, however, must be left to the judgment of the operator, and it is therefore not surprising to find that while many prefer a diagram 3in. in height, others consider $1\frac{1}{4}$in. amply sufficient. For average cases, the author favours a height of $2\frac{1}{2}$in. for low speeds, 2in. for moderate speeds, and $1\frac{1}{2}$in. for high speeds. For gas and oil engines, care must be taken to select a sufficiently strong spring, and in general a diagram-height of 2in. will be found ample. It may be noted that while the table on page 123 gives the several lists of standard springs, the various makers undertake to furnish springs for indicating other and higher pressures than those specified; while nearly all makers supply metric springs, the scale of which is expressed by the number of millimetres through which the pencil moves for each kilogramme per square centimetre of pressure.

The scale of the ordinary standard springs may be expressed in metric measurement by dividing the constant 361·2 by the number given in the first column of Table I. Thus, a standard 45 spring would be $\frac{361 \cdot 2}{45} = 8 \cdot 02$ on the metric scale. The corresponding maximum pressures can be ascertained by multiplying pressures per square inch by 14·223, thus obtaining the equivalent pressures in kilogrammes per square centimetre.

In ordering springs it should be distinctly specified whether they are for outside- or inside-spring indicators.

In changing springs, care should be taken to see that the fittings at the heads are fairly screwed home, and that no dirt or grit displaces them from their true axial position.

Springs for Gas and Oil Engine Indicators.—When standard springs are used in gas engine indicators the spring rating will be modified in accordance with the reduced area of the piston, the scale being doubled if used with a piston of half-size, and so on. A 200lb. spring used with a half-size piston, and becoming virtually a 400lb. spring, will be found sufficiently strong for general purposes, as with a diagram $1\frac{1}{2}$in. high it would cover a maximum pressure of 600lb. per square inch.

A spring of medium strength (about 50lb. ordinary rating) is useful for ascertaining the compression pressures during power strokes and to enable the point of fixing to be determined; while to observe more exactly the suction and exhaust strokes *weak-spring* diagrams are necessary. For these a spring of about 4 or 5lb. (ordinary rating) will be found suitable, but the spring must be protected from over-compression, as previously described (page 124). Messrs. Dobbie McInnes provide a special form of weak spring in which the heads are prolonged within the coil of wire so that compression is resisted while a sensitive exhaust or induction diagram is obtained. If desired, they can be made to indicate 5lb. or so on the pressure side.

Indicator Scales.—For the purpose of measuring the pressures recorded in the diagram, scales are supplied by the instrument makers for the various standard springs. Occasionally these are of steel, with a number of graduations marked on each, but more generally they are of boxwood, and as these have only one graduation on each edge, they are less liable to be wrongly applied. The steel scales are doubtless the more accurate, as they are free from liability to warp, and are less affected by differences of temperature. Scales of card, paper, vulcanite, celluloid, etc., are not to be recommended. For very accurate work, many of the makers will supply special scales made to correspond exactly to the tested deflections of the particular spring used.

Position of the Indicator.—Whenever possible, the arm or bracket carrying the paper drum should be arranged to point as nearly as possible in the direction of the motion pin of the reducing gear. A departure from this general rule may sometimes be necessary in indicating some forms of gas and oil engines, as in this case the drum must be kept as far away from the chimney of the ignition tube as possible. In this connection it may be noted that it is desirable to place the instrument so that the handle of the cock can be operated with one hand, while the other can rest upon the gas valve spindle. With the coupling used on the Richards indicator care must be taken to see that the threads lock fairly, and that it is screwed well home so as to ensure rigidity. In general, it will be found advisable to remove the pencil movement of the instrument (covering

the cylinder with a cork or piece of waste) while it is being fixed and the drum motion adjusted. With the expert operator, however, this precaution will be unnecessary.

Cord Adjustment.—The length of the driving cord must be adjusted so as to cause the drum to move midway between the two stops which limit its motion, but without coming in contact with either at the ends of the stroke. With care the required length may be readily found by bringing the cord from the paper drum, passing it round the motion pin of the reducing gear while the engine is running, and drawing it slowly round the pin until the back stop just ceases to limit the motion in that direction. Noting the point of contact of the cord and pin, the cord is drawn round the latter until the forward stop comes into play. Midway between the two positions given on the cord is evidently the point of attachment sought, and a loop should therefore be tied in the cord so that this point is in contact with the pin of the reducing gear. The loop must in all cases be sufficiently large to allow the pin to turn in it quite freely. It is scarcely necessary to caution the operator against hooking the cord on to the pin until it has been held in the position it would then occupy and the length found to be correct. In all cases it is best to err on the safe side, as too long a cord can be rectified by tying knots in it, or by slightly adjusting the position of the indicator until the drum ceases to strike the back stop. But with too short a cord the instrument may easily be severely strained or the reducing gear distorted.

New indicators have usually a short piece of cord attached to the paper drum and furnished with a light wire hook. As previously mentioned, this is intended to engage with a loop formed in the main driving cord, and if used it should be kept as close to the instrument as possible, not only for convenience in handling, but in order to prevent vibration of the cord, which is particularly liable to occur at high speeds. The same point is to be observed with the various forms of cord adjusters, some of which are sufficiently heavy to cause sagging of a fairly long cord if used near the centre of its length. These devices are made in a variety of forms, probably the oldest of which consists of a short piece of sheet brass (or aluminium) about $1\frac{1}{4}$in. in length, $\frac{1}{4}$in. in

width, and provided with four holes about $\frac{1}{4}$in. apart, as shown in Fig. 83. The cord A from the reducing gear is laced through the first two holes, and, allowing sufficient to form a loop, it is then passed up through the *fourth* hole and through the third, the end B being passed under the loop so formed. If desired, this loose end may then be led away and tied to the indicator, leaving sufficient length to prevent it pulling tight when at the extreme limit of motion; this prevents the cord falling when the hook is removed from the loop. As previously stated, with high speeds it is preferable to attach one end of an elastic band to B, securing the other end of it to the instrument or some convenient

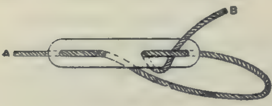

FIG. 83.

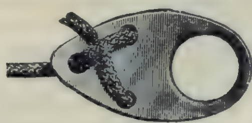

FIG. 86.

FIG. 84.

FIG. 85.

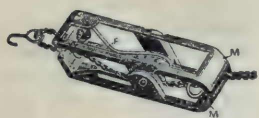

FIG. 87.

support near it in order to prevent the cord sagging when disengaged.

In place of the brass slip, a piece of leather about $1\frac{3}{4}$ in. long, $\frac{1}{2}$in. and $\frac{3}{16}$in. thick, is sometimes used, but in either case the length of the cord can be readily adjusted by sliding the buckle along the cord, the pull of the drum spring causing it to remain in position. The modified forms of cord adjusters shown in Figs. 84, 85, and 86 do not require further explanation. That shown in Fig. 87 is made by Messrs. Dobbie McInnes, Limited, and forms a convenient arrangement for the purpose. The two jaws M, M are caused to grip the cord by the action of the wire spring P forcing their rear ends apart. By pressing the latter, the length of the cord loop may be readily adjusted as desired.

Some little practice will be required before the beginner will be able to quickly engage and disengage the hook from the loop in the cord, especially at high speeds. When the elastic band is used, however, the operation is quite readily effected by catching the hook and holding it stationary so as to cause the drum to remain against the forward stop, when the loop will readily detach itself. Many modern instruments are provided with ratchet teeth at the base of the paper drum, with which a detent may be made to engage when it is desired to stop the drum (see page 20). This is a very convenient arrangement, for in many situations it is difficult to obtain ready access to the cord to unhook it.

The cord, after passing round the groove at the base of the paper drum, is passed through a small hole therein, and a knot tied on the inside. Usually the width of the cord groove is sufficient to take in two or more coils of the cord, but in practice it will be found inadvisable to use more than enough to give about one and a half turns round the drum. Nothing is gained by using a greater length than this, while, on the other hand, there is a risk of the last wrap of cord riding on the others at some part of the stroke, thus rendering the diagram utterly useless. In some modern indicators a series of holes are provided in the drum, enabling the cord to be so attached as to avoid overlapping.

Drum Spring Tension.—As previously pointed out, it will generally be necessary to increase the tension of the drum spring as the speed is increased, in order to neutralise the effect of the inertia of the drum, which would otherwise overrun the spring, causing the length of the diagram to exceed that due to the reducing motion. But in all cases the tension used should only be sufficient to keep the drum well under the control of the driving cord. With the same object in view, the travel of the paper drum—*i.e.*, the length of the diagram—is usually reduced as the speed increases, the extent to which the tension is increased, and the length of stroke diminished, depending upon circumstances. In some instances it will be found preferable to increase the drum spring tension at high speeds, in order to obtain a fairly long diagram. Generally, however, it is advisable to

keep the tension as low as convenient, and to diminish the length of the card, more especially as the height of the diagram is reduced at the higher speed owing to the use of a stronger spring. In this way a fairly proportioned diagram will be obtained.

Length of Diagram.—At low speeds, well-defined diagrams 4½in. long can be obtained by most of the standard instruments, the principal exception being the Crosby indicator. The latter instrument is, however, also supplied fitted with a 2in. drum in place of the 1½in. drum used in the standard indicator, and with this, full-length diagrams may be readily obtained. As the speeds increase, the length of the diagrams should be reduced somewhat in the following ratio :—

Speeds up to 100 revs. per minute					...4½in.	long.
,,	from 100 to	200 revs. per minute			...4	,,
,,	,, 200 ,,	300	,,	,,	...3½	,,
,,	,, 300 ,,	400	,,	,,	...3	,,
,,	,, 400 ,,	500	,,	,,	...2½	,,
,,	,, 500 ,,	600	,,	,,	...2	,,
,,	,, 600 ,,	800	,,	,,	...1½	,,
,,	,, 800 ,,	1000	,,	,,	...1¼	,,

Many prefer to use shorter diagrams for the lower speeds, restricting the length to 3 or 3½in. It will be found that a length equal to about twice the height will give a well-proportioned diagram.

Paper and Pencil.—Opinion appears to be fairly evenly divided as to the relative merits of the special "metallic" paper and the ordinary tough smooth paper which is frequently used for the purpose. With the former, the pencil should consist of a piece of brass wire brought to a fine point, yet not sufficiently sharp to cut into the paper. Gun-metal will, however, leave a more legible mark, but it is not so readily procurable as brass wire, and the latter is almost invariably used. The friction of a properly sharpened metal marking-point on chemically prepared paper is not likely to be less than with a hard lead pencil on ordinary paper, since a somewhat greater pressure is needed to make a legible line. But the point will not require one-tenth the attention that a lead will need in order to obtain a series of

distinct cards. Further, diagrams so drawn do not so readily lend themselves to "touching-up" as do those drawn with a lead pencil. On the other hand, diagrams drawn on metallic paper become indistinct after a time, and if the records are to be preserved, it is necessary to trace over the outline in ink or pencil, a tedious process when a number of diagrams have to be dealt with, and one, moreover, by which errors may be easily introduced. When a pencil of hard black lead is used, any smooth paper may be employed, and this is often a great convenience. Frequent sharpening of the lead will be necessary, however, as the fine point is rapidly worn away, and a broad and uncertain line produced. A small file will be found of service in bringing the lead to the fine round point which is necessary to secure the best results, but no greater length of lead should be used than is necessary to secure it firmly in the holder. In particular, the length of lead between the pencil lever and the paper should be kept as short as possible. With large diagrams, and in cases where it is desirable to follow the outline several times, the metallic paper and marking point will undoubtedly give better results than lead; but for single cards from high-speed engines there is little to choose between the two. In any case, the matter is largely one of convenience and individual preference.

Pads of printed blank cards with spaces provided at the top for the data necessary to be recorded not infrequently accompany new instruments, and in order to avoid encroaching on the working area of the card, these blanks are invariably made of a greater width than necessary to cover the barrel. This surplus paper does not by any means assist the beginner in mounting the paper, while it is also a disadvantage in other respects. A much better arrangement is to provide spaces for data at the *ends* of the card or on the back. Generally it is a mistake to use paper of such a width as will leave it projecting beyond the top of the drum.

If ordinary paper is used, a smooth, tough, and fairly stout variety should be selected if possible, so that while it will bear handling, the pencil will pass over it with but little friction. The paper should be cleanly cut into strips of a width about $\frac{1}{4}$in. less than the available height of the

drum, and about 1in. longer than will suffice to completely encircle the latter.

Some little practice is required in order to acquire dexterity in mounting the paper on the drum so as to avoid soiling or creasing the "card." In most of the more recent instruments, the two clips by which the ends of the card are secured are made slightly unequal in length, and this very materially assists the operator in "papering the drum." Probably the most usual method of procedure is to place the lower right-hand corner of the blank card under the top of the longer clip, after which the paper is passed

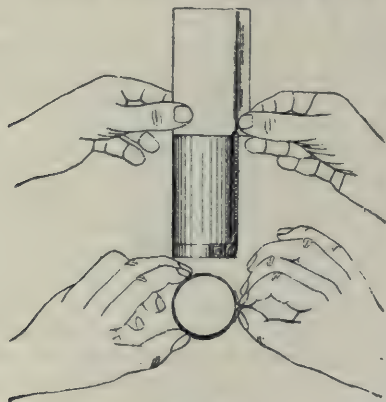

FIG. 88.

round the drum, and the other end caught by the short clip as shown in Fig. 88. The excess length of paper should be arranged to project beyond each clip equally, and by taking the lower corners between the thumb and forefinger of the right hand and aiding the operation by the left hand in the manner indicated, the paper may be drawn down neatly and tightly over the barrel (Fig. 89). Generally, however, it will be necessary to tighten the upper part by pinching the paper in near the top of the clips. The ends of the paper left projecting may either be torn off or folded back so as to lie close to the clips. If the paper barrel is removed from the instrument (which is not generally necessary), the small

finger of the left hand may be used to support it in the absence of any better expedient. If only one hand is available, the ends of the paper intended to project should be bent back beforehand. Then the paper, formed into a tube, is held bodily within the hand, the thumb and third finger gripping the outwardly-bent ends. The tube thus formed is then slipped over the top of the barrel, the projecting ends coming between the two clips. As the paper is slid down the barrel, the ends are gradually drawn farther out by straightening the finger and thumb, so that when the operation is completed the paper will fit tightly around the barrel. If thin paper is used, it will be necessary to bend the ends back close to the clips, to prevent the paper

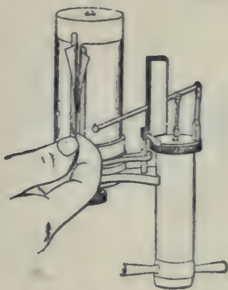

FIG. 89.

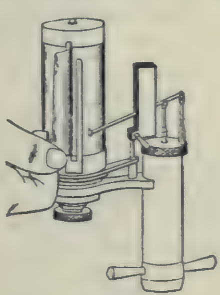

FIG. 90

slipping and becoming loose. Obviously this latter method is more practicable with small than with large barrels.

Many prefer to place the paper under both clips, as in this way all projecting ends are avoided. This implies a double thickness of paper between the clips and the barrel, and with paper of which one thickness would not give a very secure hold, it is the better plan to adopt. If thick paper is used in this way, a piece of paper packing should be placed between the lower end of the clip and the barrel, in order to increase the space between the two. With this method of papering the drum, the latter should be removed from the instrument and one end of the paper placed under both clips for about half-an-inch down. The paper is then drawn round the drum, and the overlapping end also

inserted under the clips. Then tightly encircling the lower part of the paper with the thumb and forefinger of the left hand, and holding the drum with the right, the paper is to be slipped down the barrel while the latter is slightly rotated. In another variation of this method, shown in Fig. 90, the upper layer of paper is passed under one clip only, the projecting end affording a means of drawing the paper smooth and tight.

Manipulation of the Instrument.—The instrument being in position, and the pencil pressure nicely adjusted so as to make a fine yet perfectly legible line, the driving cord should be connected and the indicator cock opened for a few strokes in order to heat up the indicator and to expel any water which may have collected in the instrument and its connections. Then turning the cock so that the small hole in its shell allows air to reach the under side of the piston, the atmospheric line is drawn. Immediately after the cock should be opened to steam and the pencil held lightly but firmly against the paper during one or more revolutions of the engine. The cock is then turned into its previous position, and the pencil again applied in order to verify the atmospheric line. Should any discrepancy appear, it will be necessary before proceeding further to disconnect the instrument and ascertain the cause. Some operators prefer to draw the atmospheric line by disconnecting the driving cord and rotating the drum by hand, as the line drawn in this way is longer than the diagram, and this avoids confusion. The fact should be borne in mind, however, that it is particularly necessary that the atmospheric line be drawn immediately after the diagram is taken, and before the instrument has had time to cool in the least degree. Therefore, unless the driving cord can be readily disconnected, it is preferable to draw the atmospheric line in the ordinary way.

If the diagram is required to show the action of the valves, one complete revolution will usually suffice; but if a power computation is to be made, the pencil should be allowed to remain in contact with the paper during some twelve or fifteen revolutions of the engine, and the mean pressure measured from the average diagram of the series which will be traced, if the load varies or if the engine runs

irregularly owing to imperfect governing, too light a fly-wheel, etc. The pencil is then withdrawn, the driving cord disconnected (or the detent used if one is provided), and the card carefully removed from the drum.

It has been previously pointed out that for exact results two indicators should be used, and diagrams taken simultaneously from both ends of the cylinder. In testing compound and triple-expansion engines, however, the simultaneous indicating of both ends of *all* the cylinders is even more desirable in order to obtain an accurate record of the action of the steam in each cylinder. It will be readily

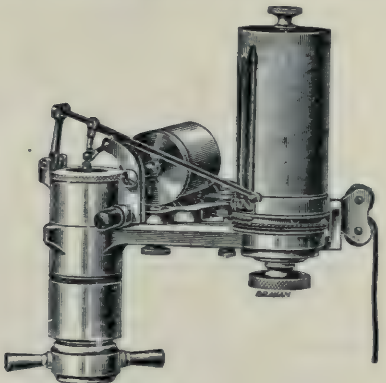

FIG. 91.

seen that this not only necessitates the employment of a number of assistants, but also some arrangement for signalling the precise instant at which the diagrams are to be taken. But even with the most careful operators errors are liable to occur in this way, and the need of some more certain and less complicated arrangement became evident as the accurate testing of multi-cylinder engines became more general. This requirement has been met by the Crosby Steam Gage Company by the introduction of Sargent's electrical attachment, which ensures the simultaneous operation of any number of indicators at any instant desired. The controlling indicator manipulated by the

operator is fitted with a simple form of circuit-closing device, so arranged that when the pencil is pressed home to take a diagram, an electrical circuit is completed, and the other instruments connected are simultaneously operated. This is effected by attaching an electro-magnet to the bracket of the indicator (Fig. 91), the armature of which is coupled to the fulcrum arm by a small latch or hook, which may be readily disconnected when it is desired to withdraw the pencil arm completely away from the drum when changing paper, etc. The movement of the armature towards the magnet is resisted by a spiral spring (the tension of which may be readily adjusted), so that normally the pencil remains at some little distance from the paper. But immediately the circuit is completed by the operation of the controlling instrument, the armature and the connected pencil movement are instantly moved and the diagram taken. Any number of instruments may be simultaneously actuated in this way. Other instrument makers furnish somewhat similar attachments. while others advocate pneumatic operating devices for the purpose. The action of the latter, however, would appear to be less decisive and reliable than with electrically operated arrangements.

In taking diagrams from *gas* engines care should be taken to avoid opening the cock at the instant an explosion occurs, as the sudden shock to which the instrument is thus subjected may cause serious derangement of the pencil movement. If the instrument is attached to the combustion chamber in such a position that one hand can be placed on the gas valve spindle, any accident of this kind may be prevented, for in this way the operator can ascertain when the gas valve opens, and by immediately pressing the pencil home, the following explosion can be safely taken. If the engine is working at full power, subsequent working strokes will retrace diagrams differing but little from that first taken. On the other hand, if the engine is running below full power, the strength of the explosive charge will vary from time to time, and a series of diagrams will be produced. For power tests, the pencil should be kept in contact with the paper for several strokes, and the mean effective pressure deduced from the average diagram.

Data to be Recorded.—The notes and particulars which

it is necessary to record upon the diagram will depend upon the nature of the test being made, but in general it will be found desirable to fill in the following details :—Date and time at which the card was taken ; the name and address of owners of engine, name of mill, etc. ; the number of the card, if one of a series ; type of engine ; which engine (R.H. or L.H.) ; which cylinder ; which end ; boiler pressure ; revolutions per minute ; length of stroke ; diameter of cylinder ; whether steam-jacketed, and how ; diameter of piston rod ; scale of spring ; vacuum per gauge in inches ; whether jet or surface-condensing ; temperature of injection water ; of hot well ; type of valve gear ; character and amount of the load, and any other notes and remarks which may appear desirable to record. For more elaborate tests, particulars will be required as to the atmospheric pressure ; pressure in the receiver ; diameter and stroke of the air pump ; whether double or single-acting ; percentage of stroke at which cut-off occurs ; clearance volume expressed as a percentage of the piston displacement ; quality of the steam ; diameter and length of steam pipe ; whether protected ; diameter and length of exhaust pipe, etc. Particulars of the boilers will also be necessary ; whether fitted with fuel economiser ; temperature of the chimney gases ; weight and temperature of the feed water used per hour ; of coal per hour, etc.

In indicating *marine engines*, the tabulated data should also include as many of the following additional particulars as the character of the test may suggest as necessary :—Name of vessel ; on what voyage ; draught forward and aft ; direction and force of the wind ; kind and direction of the sea ; speed in knots ; diameter and pitch of propeller and number of blades ; or dimensions of paddle-wheels ; position of link ; condenser surface in square feet, etc.

In indicating *locomotives*, the circumference of the driving wheel (found by carefully measuring the distance travelled during one complete revolution) ; the position of the link ; weight of train ; gradient in feet per mile ; radii of curves ; condition of the metals ; and the size of the blast-pipe orifice, cover the principal additional items requiring tabulation.

In indicating *gas engines*, note should be taken of the number of explosions per minute ; gas consumption in cubic

feet per explosion ; ignition gas consumed ; pressure of gas at meter in inches of water ; quantity of jacket water used per explosion, and the initial and discharge temperatures of the jacket water.

In indicating *air compressors*, the additional data to be noted include the barometric pressure ; pressure at the intercooler ; pressure at receiver ; quantity of circulating water ; and the temperatures of the atmosphere, of the air inlets and outlets, and of the circulating water inlet and outlet.

Concluding Notes.—When all the diagrams have been taken, the driving cord should be disconnected, the piston, spring, etc., removed from the cylinder, and steam allowed to blow through the latter for a few seconds. The spring should be removed from the piston and thoroughly cleaned and oiled in order to prevent rusting. The piston, pencil movement, etc., should also be carefully attended to, and after being thoroughly dried by a soft cloth or tissue paper, they may be slightly oiled. The body of the instrument having been removed from the engine cylinder, cleaned and oiled, the indicator should be put together and placed in the instrument box ready for future use. The springs should be screwed on to the studs usually provided in the box, and the latter locked and placed safely away in a dry place. Spare springs may be wrapped in oiled tissue paper.

The modern indicator is the outcome of many years' constant endeavour to improve the design and construction of the instrument, and with the present forms there appears but little opportunity for further advances. But the fact must not be overlooked that the later types of indicators need greater care in their manipulation and general treatment than was necessary with the older and less sensitive instruments. Intelligently used and well cared for, the modern instruments will give very satisfactory results in working, while, on the other hand, careless treatment and neglect will much more than nullify all the advantages due to excellence of design and workmanship.

APPENDIX.

RECORDING INDICATORS.

IN measuring the area of an indicator diagram by the planimeter, it is obvious that the tracing point of the latter instrument virtually retraces the figure described by the indicator pencil in combination with the movement of the paper drum. This consideration has led to the introduction of a type of indicator in which integrating mechanism is provided, this being so arranged as to record automatically the sum of the areas of any number of indicator diagrams. Various designs have been introduced from time to time, but hitherto they have not found favour, probably by reason of the somewhat complicated and delicate mechanism employed in their construction. The Bottcher recording indicator is a recent instrument of this kind, but of a more robust construction than usual. Referring to Fig. 92, it will be seen that the indicator is of the external-spring type. The top of the piston rod is fitted with a cross-head *f*, from which two links *g g* connect to the upper arms of a pair of bell-crank levers *h*, fulcrumed on the bracket *c*. The lower arms of the bell-crank levers are connected to the frame of the planimeter, which carries the meter case *o* and the measuring wheel *k*. The latter runs on the top, *r*, of the supplementary drum *s*, which is geared to the paper drum as shown. A spring *q* is provided to maintain contact between the measuring wheel and the top of the drum, and the wheel is connected to a train of gears in the meter case.

It will be seen that the movement of the piston rod causes the measuring wheel to be moved radially across the top of the drum, the combined effect of this movement and the rotation of the measuring wheel by the drum *s* being transmitted to the dials of the meter. The number

thus recorded, when multiplied by the constant for the instrument, gives the total area of the series of diagrams. In this way a continuous record of the indicated horse-power can be obtained, while the usual diagrams can be taken when desired without interfering in any way with the recording side of the instrument.

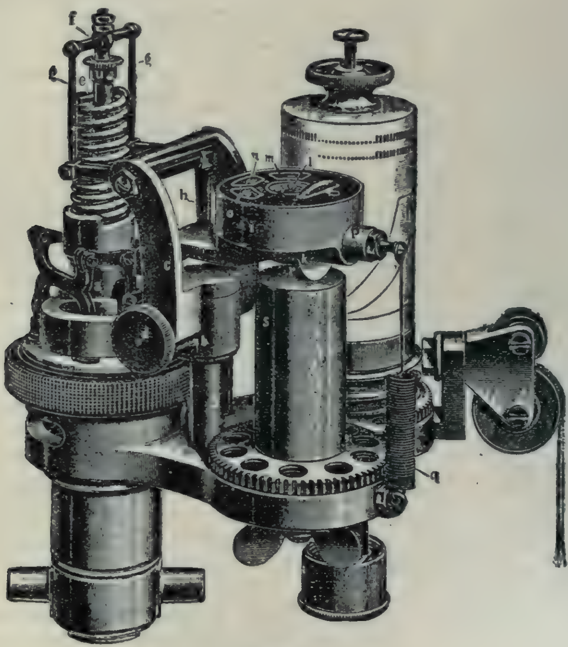

FIG. 92.

In another form of this indicator the measuring wheel runs on the top of the paper drum. This instrument has the advantage of simplicity of construction, but the paper cannot be changed without stopping the recording apparatus, and hence only a single diagram can be taken during the test.

INDEX TO PART I.

THE INDICATOR HANDBOOK.

PART II.

The Indicator Diagram: Its Analysis and Calculation.

CHAPTER I.

PRELIMINARY CONSIDERATIONS AND DEFINITIONS.

IN the initial chapter of the preceding volume of this work a rudimentary description was given of the manner in which the combined movements of the pencil and drum of the indicator result in the formation of the indicator diagram. Some slight recapitulation of the points therein advanced will be necessary in the present and succeeding sections, in which it is proposed to examine the several influences which in practical working modify the ideal diagram in a more or less marked degree.

The most elementary form of indicator diagram consists of a rectangle Z B C Z′ (Fig. 1), in which Z Z′ is the zero line or vacuum line—*i.e.* the line of no pressure,—and A L the atmospheric line, Z A representing the atmospheric pressure on the scale of the diagram. Commencing at Z, the vertical line Z B indicates the instantaneous attainment of a pressure in the cylinder, the (absolute) amount of which is measured by the height of B above Z Z′. The piston now moving to the right, and the pressure behind it

being uniformly maintained, results in the formation of the horizontal line B C, while when C, the end of the stroke, is reached, the vertical line C Z′ marks the instantaneous fall of pressure to zero. The direction of the piston is then reversed and the line Z′ Z described indicating a total absence of pressure in the cylinder during the return stroke. This completes the diagram and gives the area Z B C Z′ as a measure of the work done by the steam during the stroke.

In order to effect the instantaneous reduction of pressure indicated by the line C Z′, it may for the present purpose be supposed that means are provided by which the steam in the cylinder at the end of the stroke may be instantly

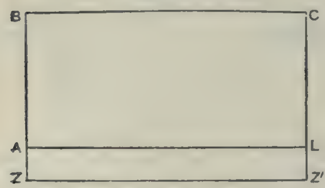

Fig. 1.

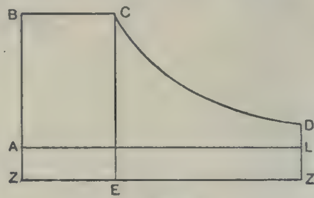

Fig. 2.

condensed. Such conditions as these may be approximated to in slow-moving condensing engines; but in non-condensing engines, where the steam is discharged into the atmosphere, the diagram is modified considerably.

In this case, steam of an absolute initial pressure Z B is admitted to the cylinder, and the line B C traced as before. At C the cylinder is placed in communication with the atmosphere, and the pressure falls until it is equal to that outside the cylinder—that is, to Z′ L. During the return stroke, therefore, the motion of the piston is opposed by a uniform pressure Z′ L equal to that of the atmosphere, and the line L A, completing the diagram, is produced. At A the pressure is increased from Z A to Z B, and the cycle of operations repeated. In this case it will be seen that the area A L Z′ Z represents work done by the steam in moving the piston against the atmospheric pressure. Accordingly this must be deducted from the area Z B C Z′

in order to obtain the area A B C L representing the amount of *effective* work performed during the stroke.

Expansion.—The distinguishing property of gases is their capability of infinite expansion, the increase of volume being accompanied by a corresponding decrease of pressure. If, therefore, gas (or steam) of a pressure Z B (Fig. 2) be allowed to enter the cylinder during a portion of the stroke Z E only, it follows that during the remainder of the stroke, E Z′, the volume of the gas will continue to increase, while the pressure will gradually fall. This action is represented by the shape of the diagram, in which Z B marks the instantaneous admission at the commencement of the stroke, and B C indicates the portion of the stroke during which the pressure is uniformly maintained. At C, however, the admission ceases, and during the remainder of the stroke the pressure gradually falls, as shown by the curved line C D. At the latter point the working fluid is discharged, and the pressure falls to that of the back pressure Z′ L, the line L A being then traced as in the previous case.

Isothermal Expansion.—The relation between the pressure and volume of a *perfect gas* is enunciated by Boyle's law :— The pressure of a gas varies inversely as the space it occupies, the temperature being constant. Or, if p = pressure, and v = volume, then $p\,v$ = a constant. From this it follows that the pressure p_1 of a gas, after expanding to a volume v_1, can be determined by dividing the product of the original pressure p and volume v, by v_1; or,

$$p_1 = \frac{p\,v}{v_1}.$$

A graphic representation of this law is given in Fig. 3, in which a quantity of gas of a pressure Z B and volume Z E is allowed to expand at constant temperature. Pressures being measured on Z B and volumes on Z Z′, it follows that if the volume be increased to Z E_2 = 2 Z E, the pressure will be diminished to $\frac{1}{2}$ Z B = Z K. By drawing the volume line K c_2 to meet the pressure line $E_2\,c_2$, the point c_2 is obtained in the expansion curve—representing the position of the pencil of the indicator under the conditions here assumed. Similarly, a threefold expan-

sion to the volume Z E_3 reduces the pressure to $\frac{1}{3}$ Z B, represented by Z N or E_3 c_3, and giving c_3 as another point in the expansion curve. Proceeding in this way, the pressures at successive volumes or piston positions E_4, E_5, etc., may be found, and through the series of points so obtained the complete curve may be drawn as shown.

From what has been said, it will be seen that every point in the curve C c_5 is so located that the product of the lengths of the perpendiculars from the points to the lines Z B and Z Z′ is constant. The curve C c_5 is therefore a

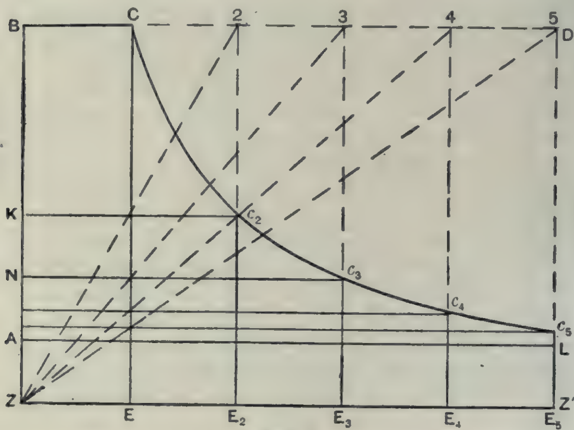

FIG. 3.

rectangular hyperbola, of which Z B and Z Z′ are its "asymptotes." Hence, the area E C c_5 E_5, representing the work done during expansion, may be determined by means of the established connection between hyperbolic superfices and their base lines.

The law of expansion above enunciated is accurately applicable only to a perfect gas expanding at constant temperature—a stipulation indicated by the term "isothermal," which is applied to expansion under these conditions. Absolutely perfect gases which rigidly follow this law are, however, unknown, although many of the so-

called permanent gases may be considered practically perfect, since they approximate very closely indeed to the ideal conception. On the other hand, gases which even remotely approach their liquefying point fail to obey the hyperbolic law of expansion, the deviation from the theoretical curve increasing as the point of condensation is approached. In the case of steam, therefore, which is eminently a condensible gas, the hyperbolic law of expansion would appear to be totally inadmissible, and in point of fact steam subjected to tests under the same conditions as a permanent gas is found to differ very materially in its behaviour during expansion, the isothermal curve deviating considerably from the hyperbola. In many cases, however, the conditions under which steam is used conspire to produce an expansion curve which closely approximates to the hyperbola. Further, the hyperbolic expansion curve has the advantage of being readily constructed, and hence it is much used both for the purpose of theoretical reasoning and as a standard with which to compare the actual expansion curves of indicator diagrams. A method of constructing the curve is shown in Fig. 3.

Disregarding clearance, the rectangle Z B C E represents the product of the initial volume (Z E) and initial pressure (Z B), C being the point at which the steam is cut off. The steam line B C is prolonged to D, and in C D a number of convenient points are selected, from each of which lines are drawn to Z. From the points in which these lines cross the cut-off ordinate C E, horizontals are drawn intersecting a series of vertical lines which are let fall from the points 2, 3, 4, etc. Through these points of intersection c_2, c_3, etc., the expansion curve is drawn.

Expansion Curves for Saturated Steam: The Saturation Curve.—When dry, saturated steam expands, doing external work, and sufficient heat is supplied to just prevent liquefaction (that is, to keep the steam *saturated**), the pressure at any point in the stroke is that due to the volume and temperature of saturated steam. Hence the resulting curve is frequently termed the *saturation curve.* Since con-

* The term designating steam in that condition in which the slightest increase of volume or reduction of temperature will cause a portion to liquefy.

densation is only just prevented, the pressure falls below that due to isothermal expansion, the temperature falling in correspondence with the decreasing pressure. Rankine has established an approximate formula for cases of this kind, which is applicable for pressures of from 30 to 120lb. per square inch, and for ratios of expansion of from 4 to 16. In this the relation of the pressure and volume is given by the expression $p\ v^{\frac{17}{16}}$ = a constant. Dry saturated steam expanding in a well-jacketed cylinder may be expected to yield an expansion curve of this class.

The Adiabatic Expansion Curve.—When saturated steam expands, doing external work in a non-conducting cylinder, the fall of pressure which ensues is due both to the increase of volume and the liquefaction of a portion of the steam, owing to the disappearance of the heat equivalent of the external work done. Expansion under these conditions, which has been designated "adiabatic," is to be regarded as an ideal conception, impossible of being realised, since in practice non-conducting cylinders are non-existent. Occasionally, however, the curve is approximated to in diagrams from well-lagged cylinders or high-speed engines. With expansion under these conditions the pressure decreases more rapidly than in the previous case. Rankine's rule connecting the pressure and volume under these conditions, and applicable to pressures of from 15 to 180lb. per square inch, is given by the expression $p\ v^{\frac{10}{9}}$ = a constant.

Other Expansion Curves.—Other empirical rules connecting the pressure and volume of expanding steam are sometimes used as more nearly representing the actual conditions than either of the foregoing. Thus, for superheated steam $p\ v^{1 \cdot 3}$ = a constant, and for very damp steam, $p\ v^{\frac{15}{16}}$, have been employed, while occasionally the expansion is assumed to be represented by such constants as $p\ v^{\frac{7}{8}}$ and $p\ v^{\frac{11}{10}}$. It will be noted that the indices or exponents in all these expressions may be presented in a slightly different form. Thus $p\ v^{\frac{17}{16}}$ may be written $p\ v^{1 \cdot 0625}$, the exponent here being the result of dividing 17 by 16. For saturated steam, Zeuner uses an expression of this kind ($p\ v^{1 \cdot 0646}$) as being more nearly correct than Rankine's $p\ v^{\frac{17}{16}}$. In the compression of air the ratio (1·408) of its specific heat at constant pressure to that at constant volume is used for the exponent in theo-

TABLE I.—TERMINAL PRESSURE FACTORS.

Ratio of Expansion.	Saturation Curve.	Adiabatic Curve.	Ratio of Expansion.	Saturation Curve.	Adiabatic Curve.	Ratio of Expansion.	Saturation Curve.	Adiabatic Curve.	Ratio of Expansion.	Saturation Curve.	Adiabatic Curve.
1·0	1·0	1·0	4·6	0·197	0·183	8·2	0·107	0·096	14·6	0·0579	0·0508
1·1	0·900	0·900	4·7	0·193	0·179	8·3	0·105	0·095	14·8	0·0571	0·0501
1·2	0·826	0·819	4·8	0·189	0·175	8·4	0·104	0·094	15·0	0·0563	0·0493
1·3	0·757	0·746	4·9	0·184	0·171	8·5	0·102	0·092	15·2	0·0555	0·0486
1·4	0·699	0·689	5·0	0·180	0·167	8·6	0·101	0·091	15·4	0·0547	0·0479
1·5	0·649	0·637	5·1	0·177	0·163	8·7	0·100	0·090	15·6	0·0540	0·0472
1·6	0·606	0·591	5·2	0·173	0·160	8·8	0·099	0·089	15·8	0·0533	0·0465
1·7	0·568	0·555	5·3	0·170	0·156	8·9	0·098	0·088	16·0	0·0525	0·0459
1·8	0·534	0·520	5·4	0·166	0·153	9·0	0·096	0·087	16·2	0·0518	0·0453
1·9	0·505	0·490	5·5	0·163	0·150	9·2	0·094	0·084	16·4	0·0512	0·0446
2·0	0·478	0·463	5·6	0·160	0·147	9·4	0·092	0·082	16·6	0·0505	0·0441
2·1	0·454	0·438	5·7	0·157	0·144	9·6	0·090	0·080	16·8	0·0499	0·0434
2·2	0·432	0·416	5·8	0·154	0·141	9·8	0·088	0·079	17·0	0·0492	0·0429
2·3	0·413	0·396	5·9	0·151	0·139	10·0	0·0865	0·0772	17·2	0·0486	0·0423
2·4	0·395	0·378	6·0	0·149	0·136	10·2	0·0848	0·0755	17·4	0·0480	0·0418
2·5	0·378	0·362	6·1	0·146	0·134	10·4	0·0830	0·0739	17·6	0·0475	0·0413
2·6	0·362	0·346	6·2	0·143	0·131	10·6	0·0814	0·0724	17·8	0·0469	0·0408
2·7	0·348	0·332	6·3	0·141	0·129	10·8	0·0798	0·0710	18·0	0·0463	0·0403
2·8	0·334	0·318	6·4	0·139	0·127	11·0	0·0782	0·0696	18·2	0·0458	0·0398
2·9	0·322	0·306	6·5	0·137	0·125	11·2	0·0768	0·0682	18·4	0·0453	0·0393
3·0	0·311	0·294	6·6	0·134	0·123	11·4	0·0753	0·0669	18·6	0·0447	0·0388
3·1	0·301	0·285	6·7	0·132	0·121	11·6	0·0739	0·0656	18·8	0·0443	0·0384
3·2	0·290	0·274	6·8	0·130	0·119	11·8	0·0726	0·0644	19·0	0·0437	0·0379
3·3	0·281	0·265	6·9	0·128	0·117	12·0	0·0713	0·0632	19·2	0·0432	0·0374
3·4	0·272	0·257	7·0	0·126	0·115	12·2	0·0701	0·0620	19·4	0·0428	0·0371
3·5	0·263	0·248	7·1	0·124	0·113	12·4	0·0689	0·0609	19·6	0·0423	0·0366
3·6	0·256	0·241	7·2	0·122	0·111	12·6	0·0677	0·0598	19·8	0·0419	0·0362
3·7	0·249	0·233	7·3	0·120	0·109	12·8	0·0666	0·0588	20·0	0·0414	0·0358
3·8	0·242	0·226	7·4	0·119	0·108	13·0	0·0655	0·0578	21·0	0·0393	0·0339
3·9	0·235	0·220	7·5	0·117	0·106	13·2	0·0644	0·0568	22·0	0·0374	0·0322
4·0	0·229	0·214	7·6	0·116	0·105	13·4	0·0634	0·0559	23·0	0·0357	0·0306
4·1	0·223	0·208	7·7	0·114	0·103	13·6	0·0624	0·0550	24·0	0·0341	0·0292
4·2	0·217	0·203	7·8	0·112	0·101	13·8	0·0615	0·0541	25·0	0·0326	0·0279
4·3	0·212	0·198	7·9	0·111	0·100	14·0	0·0605	0·0533	26·0	0·0313	0·0267
4·4	0·207	0·193	8·0	0·109	0·099	14·2	0·0596	0·0524	28·0	0·0290	0·0246
4·5	0·202	0·188	8·1	0·108	0·097	14·4	0·0587	0·0516	30·0	0·0269	0·0228

retical investigations, while in practical work $n = 1{\cdot}2$ is often used when the air is cooled by water injection, and $n = 1{\cdot}3$ when a water jacket is used.

Gas-engine Expansion and Compression Curves.—The theoretical expansion curve given by the explosion of a gaseous mixture in the cylinder of a gas engine is an adiabatic curve in which the value of n in $p\ v^n = \text{constant}$ is the ratio of the specific heats of the products of combustion at constant pressure and constant volume. The value of n will therefore vary with the proportions of air and gas forming the explosive charge, increasing from about 1·35 for mixtures of 4 to 1, up to about 1·39 for mixtures of 15 to 1. Having in view the fact that in practice the explosion of the charge takes place in a cylinder enveloped in a water jacket, resulting in the abstraction of a large amount of heat during expansion, a material deviation from the adiabatic curve might reasonably be looked for in actual diagrams. As a matter of fact, however, the discrepancy in an average diagram is comparatively slight—the heat lost through the jacket being probably compensated by a slight prolongation of the combustion period after the maximum pressure has been attained.

In gas engine compression curves ordinary values of n range from 1·28 to 1·35.

Construction of Theoretical Expansion Curves—The construction of the expansion curves $p\ v^{\frac{17}{16}} = \text{con.}$, and $p\ v^{\frac{10}{9}} = \text{con.}$, is very readily effected by the aid of the table of terminal pressure factors (page 15). In using the table for this purpose, a series of ordinates are drawn as in Fig. 4, the first passing through C, the point of origin of the curve. The distances E – ·2, ·2 – ·4, etc., are in this diagram taken at $\frac{1}{5}$ Z E, but any other convenient fraction of the admission volume Z E may be selected. To find the pressure at c corresponding to the terminal pressure of an expansion $\frac{Z \cdot 2}{Z\ E} = \frac{6}{5} = 1{\cdot}2$, reference is made to the table, where opposite the expansion ratio 1·2 are found pressure factors of 0·826 for the saturation curve and 0·819 for the adiabatic. Thus, if the initial pressure Z B is 100lb., the pressure ·2 c_1 or Z K will be 82·6lb. for the saturation curve and 81·9lb. for the adiabatic curve. The pressures

corresponding to expansion ratios of 1·4 (Z M), 1·6 (Z N), etc., may be similarly obtained, and then through the points C c_5 the required curve may be drawn as shown in the figure, which is plotted for adiabatic expansion.

Fig 5 shows a convenient method of obtaining points in an expansion curve represented by $p\ v^n = \text{const.}$ where n is any number whatever. From the point Z a line Z N is drawn, making any suitable angle Z′ Z N with Z Z′. From the same point Z M is drawn at an angle B Z M to B Z, such that $(1 + \tan.\ B\ Z\ M) = (1 + \tan.\ Z'\ Z\ N)^n$. Taking B C and

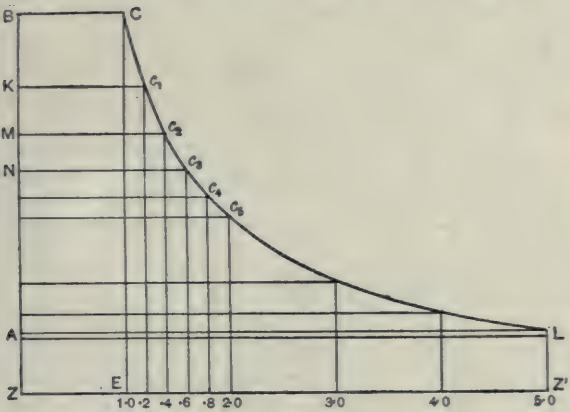

FIG. 4.

H C as representing the initial volume and pressure of the gas, B D is drawn at 45° to B Z, cutting Z M in D. D E is then drawn parallel to Z Z′ and C H produced to meet Z N in A, from which point A F is drawn at 45° to Z Z′. An ordinate E F G drawn through F cuts D E in E, which is a point in the required curve. The next point is obtained in a similar manner by drawing K L at 45° to B Z, L P parallel to Z Z′, and G J at 45° to Z Z′, thus obtaining J, through which point the ordinate J P is drawn, giving P, the second point in the curve.

Fig. 5 is drawn for $n = 1{\cdot}35$ and the angle Z Z′ N is 20°. To obtain the angle B Z N, we have—

$$(1 + \tan.\ 20^\circ)^{1 \cdot 35} = (1 + \tan.\ B Z N)$$
$$1 \cdot 364^{1 \cdot 35} = (1 + \tan.\ B Z N)$$

Working by logarithms,* we have—

$$\log.\ 1 \cdot 364 = 0 \cdot 1348 \qquad 0 \cdot 1348 \times 1 \cdot 35 = 0 \cdot 18198$$
$$\text{antilog. } 0 \cdot 18198 = 1 \cdot 521, \text{ whence } \tan.\ B Z M = 0 \cdot 521,$$
$$\text{or } B Z M = 27 \cdot 5^\circ.$$

For plotting expansion curves the author strongly advocates the use of logarithmic section paper, by the use of which the pressures corresponding to expansions from any initial pressure and for any value of n in $p\ v^n = \text{con.}$, may

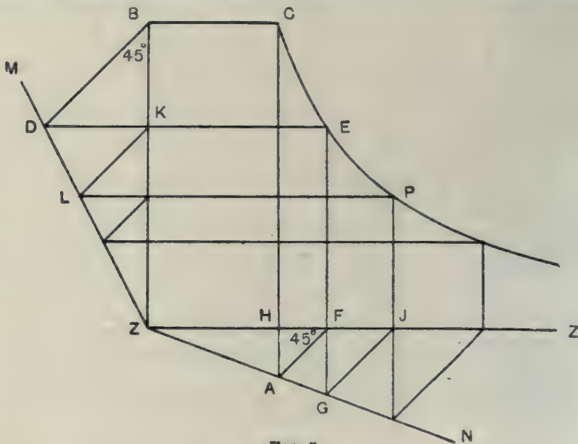

FIG. 5.

be readily determined; while, conversely, the value of n for any actual diagram may be ascertained with equal facility.

DIAGRAMS IN PRACTICE.

Thus far the indicator diagram has been regarded solely from a theoretical point of view. It now becomes necessary to ascertain the character and extent of the influences which modify the ideal diagram under the actual conditions of practical working.

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* For a simple explanation of logarithms and their application to problems of this character the reader is referred to the author's *Logarithms for Beginners* (see advertisement pages).

The influences which require to be considered in this connection are: (1) Those due to the engine mechanism and the mechanical arrangement of the motor generally; and (2) those due to thermal changes in the working fluid.* Among the principal items in the first-named section are:—Resistance of steampipes, ports and passages, valves, etc., both during admission and exhaust; cylinder clearance and compression; defective valve gear, etc. Under the second heading we have:—Cylinder condensation and re-evaporation; loss of heat by radiation and otherwise; and the means adopted to reduce these losses. But although it is convenient to differentiate between the mechanical and thermal influences which affect the practical indicator diagram, the various effects cited are so closely interwoven that it becomes preferable to deal with them generally and in the order in which they are met in tracing the course of the steam through its cycle of operations in the engine cylinder.

Fig. 6, in which the theoretical and actual diagrams are compared, will serve to indicate the character of the deviations from the theoretical conditions which are met with in practice. In the first place it is necessary to observe that in an actual cylinder, a space must necessarily be left between the cylinder cover and the piston when the latter is at the end of its stroke. The volume of this space, together with that of the passage intervening between it and the steam valve, form the *clearance volume* of the cylinder, which is conveniently expressed as a percentage or fraction of the working length of the cylinder. Thus, if in Fig. 6 the piston displacement B′ D′ is 100cub. in., and the total clearance volume for one end of the cylinder is 5cub. in. —then $B' B = \frac{5}{100}$, or $\frac{1}{20}$th of B′ D′ represents the clearance volume as a fraction of the working stroke, the line B Z being the clearance line or line of no volume.

It is evident that until C, the point of cut-off, is reached, the steam in the clearance space has no influence upon the actual work done on the piston—constituting, in fact, merely a prolongation of the steam supply pipe. As soon, however, as the steam port is closed, the volume of steam expanding

* The effects upon the diagram produced by errors of the instrument, and those due to its faulty actuation and use, have been fully discussed in the first part of this work.

behind the piston is no longer represented by B′ C, as previously assumed, but by the larger volume B C. Hence, if C D is the resulting expansion curve, the maximum work obtainable from the steam in a non-condensing engine would be represented by the diagram E B′ C D L, and under exceptional conditions such a diagram may be roughly approximated to. A typical indicator diagram, however, would show such deviations from the theoretical, as are indicated by the broken lines. Steam being admitted at *a* (the clearance already containing steam of pressure *k a*), the admission line *a b* marks the rapid (but not instantaneous) attainment of the initial pressure, which, however, falls more or less short of the boiler pressure E B′ by an amount depending upon circumstances to be discussed subsequently. From *b* to *c* a more or less horizontal *steam line* is traced,

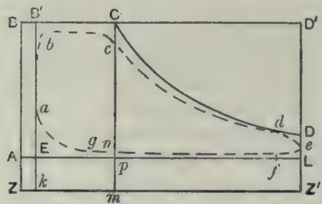

FIG. 6.

c, the *point of cut-off*, being less sharply defined than in the theoretical card, the line merging into the *expansion line c d*. Compared with the theoretical expansion curve, the line *c d* will often be found lower in the earlier part of the expansion period, while coinciding with or rising above C D during the latter part of the stroke. At *d*, the *point of release*, the discharge of the steam commences, the consequent rapid fall of pressure resulting in the formation of the *release line d e*, and leaving an absolute *final pressure* Z′ *e* in place of Z′ D in the ideal diagram. The point *e* marks the commencement of the return stroke, and the continuance of the rapid fall of pressure is then shown by the reverse curve *e f*, in the early part of the *exhaust line e f g*. The portion *f g* is often approximately parallel to the atmospheric line, its height *p n* above the latter at any point *n* representing the amount of *back pressure* which exists in the cylinder over and above

that due to the atmosphere $m\ p$, the total absolute back pressure at this point being therefore $m\ n$. At the *compression point*, or *point of exhaust closure*, g, the discharge of steam is arrested, that portion which remains in the cylinder being compressed into the clearance space by the advancing piston, and resulting in the formation of the *compression curve* $g\ a$. At a, the termination of the stroke, the *compression pressure* $k\ a$ is augmented by the admission of a fresh supply of steam, and the sequence of events recommences.

In condensing engines the cycle of operations is essentially similar, but the steam in this case discharges into a condenser in which a more or less perfect vacuum is maintained. In consequence, the release, exhaust, and compression lines are most usually found below the atmospheric line, as shown in Fig. 7. The back pressure as $m\ n$ represents in this case the pressure due to air and vapour in the

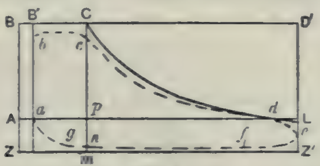

FIG. 7.

cylinder and condenser. The extent to which the back pressure falls below that of the atmosphere as $p\ n$ is frequently referred to as the *vacuum*. The obviously incorrect term, *vacuum pressure*, is sometimes used in this connection. The position of the *zero line* Z Z′ is usually decided by drawing it at a distance representing 14·7lb. per square inch below the atmospheric line A L, this being the average value of the pressure of the atmosphere. But for all exact calculations and comparisons, the height of the barometer at the time of the test should be observed, and the zero line drawn below A L at a distance equal to the corresponding *barometric pressure*. (*N.B.* 1in. of mercury = 0·49lb. per square inch.) Occasionally in diagrams from condensing engines a reference line is drawn representing by its height above Z Z′ the average pressure of the vapour in the condenser, as shown by the vacuum gauge. A comparison of

the back pressure line with this line of condenser pressure enables an estimate to be formed of the work done in expelling the steam from the cylinder.

The exhaust line of the diagram, as *e f g* (Fig. 6), is often somewhat inaccurately referred to as the *back pressure* line. It is evident, however, that the latter term applies to the whole of the diagram described during the return stroke of the piston, and hence it usually comprises a portion of the release line, the whole of the exhaust line, the whole of the compression line, and more or less of the admission line when lead is given.

Absolute pressures are pressures reckoned from a perfect vacuum, while *gauge* or *boiler pressures* are pressures above the atmosphere, and are such as are shown by an accurate steam gauge. In an indicator diagram, therefore, absolute pressures will be measured from Z Z′, and gauge pressures from the atmospheric line A L.

CHAPTER II.

THE DIAGRAM IN DETAIL.

The Admission Line.

A LOSS of pressure almost invariably occurs in the transference of the steam from the boiler to the engine cylinder, and in many cases the initial pressure realised in the latter falls considerably below that indicated by the boiler pressure gauge. The extent of this loss will depend upon a variety of circumstances. Steampipes of insufficient diameter or of considerable length, especially when they are not well clothed, result not only in a fall of pressure, but also cause the steam to be delivered in a more or less damp condition. Constricted passages, elbows, valves, and abrupt bends also occasion loss of pressure, but in this case the steam is generally slightly dried. The friction of steam passing through long pipes also tends to cause slight superheating, but in most cases this is more than neutralised by the loss due to radiation which occurs even with the most efficiently protected pipes. The loss of pressure will be greater with damp steam than with dry, by reason of its increased resistance to motion. Generally, it may be said that in the absence of any special drying arrangements, the steam reaches the cylinder in a more or less moist state. Straight, well-drained, and thoroughly-clothed steampipes, and the provision of an efficient separator, will, however, considerably improve the condition of the working steam.

The foregoing causes of loss of pressure are more particularly associated with the passage of steam from the boiler to the steamchest. The amount of the loss up to this point may be readily ascertained by applying the indicator first to

the steamchest, and immediately afterwards to the boiler, when the difference will be at once apparent. The instrument should in every case be attached to the boiler in order to obtain a trustworthy comparison, since pressure gauges are frequently incorrect in their indications.

Lead.—The precise point in the stroke at which the admission of steam commences is largely governed by the piston speed adopted. In high-speed engines it is found necessary to allow the steam to commence entering the cylinder before the piston is in a position to commence its outward stroke, the amount of this anticipatory opening—generally designated *lead* or *pre-admission*—varying considerably in different engines and under different circumstances. In engines in which little compression is possible, additional lead will assist in cushioning the reciprocating parts of the engine, but more generally the object is to ensure the prompt entry of the steam during the admission period by the early filling of the passages, ports, and clearance space generally, so that when the piston commences its outward travel, fall of pressure is in a large measure avoided. Again, since the admission steam is in contact with the piston and clearance surfaces for a longer period, the steam condensed in heating these surfaces is immediately replaced by fresh boiler steam, without causing that undue falling of pressure during admission which so frequently occurs when zero or negative lead is adopted.

Excessive lead, especially when accompanied by small compression, will frequently cause irregular running of an engine, sometimes resulting in "pounding" and heating of the bearings; but in engines having large clearance volumes, and, as already mentioned, in high-speed engines, absence of lead invariably diminishes the admission pressure realised upon the piston. When considerable compression is given, less lead will be required, since the clearance space is already filled with steam of comparatively high pressure. In all cases the prompt admission of steam is the point to be attained, but even this must, of course, be sacrificed if the necessary lead causes irregular running of the engine.

Under the best conditions of practice the admission line takes the form of a straight line rising from the termination of the compression curve and perpendicular to the atmo-

spheric line as shown at *a*, Fig. 8. Slight deviations in either direction from the perpendicular are, however, well within the limits of good practice. Cases of too early admission are indicated in *b* and *c*. In *b* the line is practically straight, but inclines towards the perpendicular to a somewhat marked degree. The form shown at *c*, sometimes met with, is curved somewhat reversely and terminates in a lip at its upper extremity. Condensation of the steam in heating up the clearance surfaces would account for this reverse curve, while the small lip *p* indicates that as practically full steam pressure is attained before the end of the stroke, the steam is momentarily forced back into the steam-chest by the advancing piston.

The cases shown in *d*, *e*, and *f* cannot properly be termed

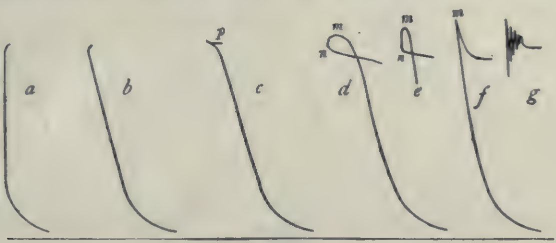

FIG. 8.

admission lines. It will be noted that in each case the apparent admission line rises materially *above the steam line*—a fact which points to excessive compression, and in no way indicates excessive lead, but rather the reverse. The port opening to steam is approximately indicated in each case by the point *m*, at which the pressure commences to fall as the compressed steam is forced back into the steam-chest, until, at *n*, the end of the stroke, the pressure is reduced to that of the admission steam. When full compression is attained early in the stroke, the resulting loop partakes of the form shown at *d*, this being modified as at *e* when the full compression is reached at a point nearer the end of the stroke. When the maximum compression pressure is reached precisely at the end of the stroke, a peak as in *f* takes the place of the loop. Care must be taken to

distinguish between peaks formed in this way and those due to the momentum of the piston and attached mechanism of the indicator. Peaks due to the latter cause are usually of a more angular character, and generally show more than one oscillation, as in *g*.

Several cases of late admission or *negative lead* are shown in Fig. 9. As in the previous examples, much depends upon the amount of compression given. With slight compression and a somewhat tardy admission, the line resembles that shown at *a*; when similar conditions obtain, but in a more marked degree, lines such as *b* and *c* result. With a fair amount of compression and the admission delayed somewhat, a fall of the compression pressure occurs, as shown at *d*, in which *m* marks the point of valve opening

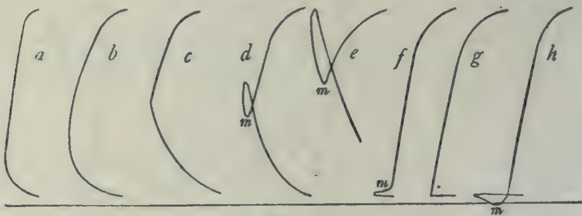

FIG. 9

The loop thus formed is occasionally accentuated to the extent shown in *e*, indicating a greater amount of compression and a further delayed admission. The form shown at *f* is obtained from engines of the Corliss and other types provided with separate admission and exhaust valves, when, owing to the exhaust valve remaining open after the end of the stroke is reached, the admission steam passes directly into the exhaust. The exhaust valve closing at *m*, the admission line is formed, being considerably inclined owing to the fact that the piston is now moving somewhat quickly. In *g* similar conditions are indicated, but the exhaust opening is much less. When the exhaust closes at the end of the stroke and negative lead is given, the fall of pressure due to the outward movement of the piston results in the formation of the loop shown at *h*, in which *m*, as before, indicates approximately the point at which admission commences.

The Steam Line.

Wiredrawing During Steam Supply.—In passing from the steamchest to the cylinder, the chief loss of pressure incurred is due to the constricted ports, and also, with the more ordinary arrangements of valve gear, to the gradual movement of the valves governing the flow of steam. For the "wiredrawing" or "initial expansion" which thus results, the obvious remedy lies in the provision of large, short, and direct steam ports and passages, together with rapidly moving valves. In the Corliss type of engine these conditions are closely fulfilled, and the result is an almost total absence of wiredrawing, steam lines of the form shown in Fig. 10 (which may be considered as the ideal form) being closely approached. In ordinary slide-valve engines regulated by governor and throttle valve, wiredrawing invariably occurs, being in this instance due to the comparatively long

Fig. 10.

Fig. 11.

Fig. 12.

and winding passages, to the slow movement of the slide-valve, and to the action of the throttle valve. In high-speed engines these effects are intensified, since there is less time allowed for the realisation of the steam pressure during admission. In many cases, however, the increased size of pipes, ports, and valves necessary to entirely obviate wiredrawing would be such as to occasion a considerable increase in condensation, valve friction, leakage, etc., as well as in the amount of power required to operate the valves, so that the apparent gain may be more than neutralised. Moreover, wiredrawing during admission tends to dry the entering steam, but in general the effect is comparatively slight.

Piston speed is an all-important factor in the wiredrawing of steam during admission, and it is evident that with a given port opening and area of piston there is a limiting speed beyond which the steam will be unable to flow intc the cylinder with sufficient celerity to maintain its initial pressure. Fig. 11 is an example of excessive wiredrawing

in a locomotive, the outer figure showing the form of steam admission line obtained with a piston speed of 300ft. per minute, while the inner lines show the effect of increasing the piston speed to 900ft. per minute, the cut-off being unaltered.

Undoubtedly the most rational manner of using steam with due regard to economy is to employ the governor to automatically vary the amount of steam admitted to the cylinder, instead of varying the pressure by more or less throttling the steam supply. By the former arrangement the initial pressure realised upon the piston approaches the boiler pressure to within 5 to 10 per cent. if a large and direct steam supply pipe is used. In diagrams from automatic expansion engines the steam line is, or should be, approximately parallel to the atmospheric line, the pressure being maintained uniformly almost to the point of cut-off. Not infrequently, however, the line falls somewhat, while slight wiredrawing almost invariably occurs during the closing of the valve, the effect being exhibited in the diagram by the dotted curve *b c*, Fig. 10, the point *b* indicating the commencement of the cut-off, and *c* the point at which the valve is completely closed. The latter is the point of contrary flexure, the curve *b c* changing into one of opposite convexity *c d* as shown. At high speeds, wiredrawing at cut-off increases considerably.

A steam line as Fig. 12 is usually obtained from an engine whose steamchest is sufficiently large to act somewhat as a receiver. Here the fall of pressure is arrested after the valve is well open, and the line may run almost horizontally until the increasing piston speed makes larger demands upon the steam supply, and the line falls as shown. This drooping of the steam line is frequently seen in diagrams showing a prolonged steam supply. Double admission, as shown in Fig. 13, may be due, with Corliss valves, to the rebounding of the dashpot, or, in engines with a sensitive shaft governor and light fly-wheel, to the sudden imposition of a heavy load causing the governor to reopen the valve. When the cut-off occurs very early in the stroke the steam line loses its distinctive character, being reduced to a mere rounded peak as in A, Fig. 14, while in diagrams taken from unloaded condensing engines (friction diagrams) the

steam admitted to the clearance space is often sufficient to drive the engine, in which case the admission line entirely disappears (B, Fig. 14).

Steamchest Diagrams.—Diagrams taken from the steamchest are frequently of value in indicating the source of wiredrawing — whether due to insufficient port area, obstructed passages, or the action of the throttle valve; insufficient size of steampipe may also be detected in this way. In applying the indicator to the steamchest, a position should be selected which is as far as possible clear of any direct current of steam. Motion should be given to the indicator barrel by precisely the same means as employed for taking the cylinder diagrams, and it is preferable that both be taken upon the same card, and as nearly simultaneously as possible. If before removing the paper from the barrel the instrument can be directly applied to the boiler, and the barrel rotated by hand so that the pencil draws the

FIG. 13.

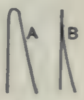

FIG. 14.

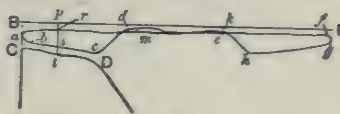

FIG. 15.

boiler pressure line, sufficient data will be available to decide the cause and extent of any loss of pressure which may occur. Steamchest diagrams vary in outline somewhat, but in general they are of the double-loop form shown in the upper part of Fig. 15. Here B P is the line of boiler pressure drawn as above described, while B a shows, on the scale of the diagram, the loss of pressure between boiler and steamchest. At a the port opens to steam and the steamchest pressure falls as shown by the line $a\,b\,c$, the portion $b\,c$ being often approximately parallel to the steam line C D of the main diagram. When the cut-off point D is reached, the steamchest pressure rises somewhat rapidly to d, the momentum due to the arrested steam flow causing the pressure to approach, and sometimes to exceed, the boiler pressure. During the remainder of the stroke the pressure is often fairly uniform, until at f the return stroke ($g\,h\,e\,m$) commences. It will be seen that $p\,r$ represents the normal loss

of pressure between the boiler and steamchest; *p s* this loss when the steam port is fully open; while *s t* represents the further loss due to the resistance of the steam port and passage. Hence a fairly close agreement of *b c* with C D, but with both sloping as shown in Fig. 16, indicates a throttling of the steam supply before it reaches the steamchest. On the other hand, in Fig. 17 but little loss is shown between boiler and steamchest, but a considerable amount between steamchest and cylinder, clearly indicating insufficient port opening.

The Expansion Line.

Under ordinary circumstances steam in a saturated condition and containing a percentage of moisture is admitted to a cylinder which, having been just previously exposed to the condenser (or to the atmosphere), may be assumed to be at

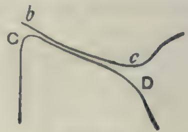

Fig. 16.

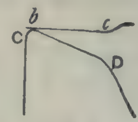

Fig. 17

a considerably lower temperature than the entering fluid. The immediate result is a partial condensation of the steam owing to a portion of the latter parting with its latent heat in order to elevate the temperature of the cylinder walls. While communication with the boiler is maintained, steam will continue to enter the cylinder, replacing that lost by condensation, and at the point of cut-off the cylinder will contain, in addition to the initial volume of steam, a quantity of water of condensation at the same temperature. As the steam now commences to expand, the pressure falls, while simultaneously the temperature of the steam is reduced somewhat by reason of the conversion of heat into work. A point is reached, therefore, at which the temperature of the cylinder walls exceeds that of the contained steam, and as a consequence a portion of the water in the cylinder will be re-evaporated. Hence, at the end of the expansion period the cylinder will contain a larger quantity of steam than at the commencement of that period by an amount depending mainly

upon the mean temperature of the cylinder walls, which in turn is influenced by the degree of expansion employed.

The effect of this condensation and re-evaporation upon the expansion line of the diagram will be to cause it to fall below the theoretical curve during the earlier portion of the stroke, and to rise above it during the latter part of the period. From this it might be inferred that under the conditions which most usually obtain, the use of the hyperbola as a curve of reference is very much open to question, and in point of fact it is impossible to base any close reasoning upon the coincidence of the expansion line with the standard curve. At the same time a comparison of this kind is often of value in detecting leakage past the valves or piston, while it is in a general way useful in considering the behaviour of steam in the cylinder.

The method of drawing the hyperbolic curve upon the diagram is readily deduced from the directions previously given (Fig. 3), but it is first necessary to decide the point of origin in the actual expansion curve through which the reference curve is to be drawn. Thus in locating the hyperbolic curve upon the diagram shown in Fig. 18, it will be understood that any point in the expansion line C *t* may be selected as the point of origin, always provided that both steam and exhaust valves are closed. In general, however, the curve is drawn through (1) a point representing the position of the piston immediately before the exhaust valve opens, as *t*; (2) a point just after the steam valve is closed, as *p*; or (3) a point midway between these two positions. The first position is selected when it is desired to show the amount of work which might have been performed by the steam accounted for by the indicator at the end of the expansion period. The second position is taken when it is desired to show to what extent the terminal pressure exceeds that due to hyperbolic expansion from the volume of steam in the cylinder at cut-off; while the third point is selected when an average comparison is desired.

Assuming the point *t* to be selected, the ordinate B E is drawn touching the diagram on the left as shown. The clearance volume E Z taken as a percentage of the piston's displacement is set off from E, and the ordinate of zero

volume Z V is drawn. Through t a horizontal line t S is drawn, and at t a vertical t D is erected. From Z a series of lines are then drawn, as shown, cutting t S in a b c and d, and t D in a' b' c' d'. Verticals drawn through the former and horizontals through the latter give a series of points through which the curve C t may be drawn, as shown. In a similar manner curves may be drawn through any other desired point in the expansion curve.

Another construction, which has the merit of simplicity, is shown in Fig. 19. Here p is the assumed point of origin, and Z Z′ and Z B the lines of zero pressure and zero volume respectively. Through p a series of straight lines are drawn, terminating in Z B and Z Z′ as shown. Then from a' on

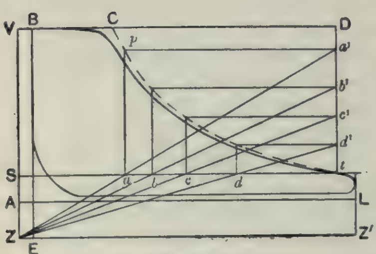

FIG. 18.

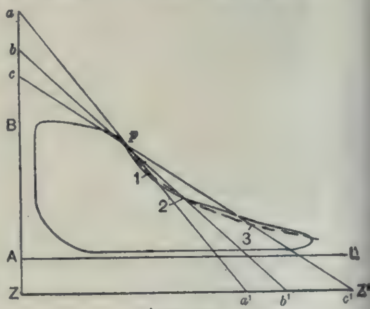

FIG. 19.

the line $a\,a'$, a length $a'\,1 = a\,p$ is marked off, giving a point in the curve. Similarly, from b' the length b' 2 is laid off equal to $b\,p$, and the curve drawn through the points 1, 2, 3, etc., so obtained.

As previously intimated, discretion must be exercised in basing conclusions upon the coincidence or otherwise of actual and theoretical expansion curves. More especially is this the case in considering diagrams from unjacketed cylinders and with high ratios of expansion. When, however, the cut-off occurs at or later than half-stroke, and fairly dry steam is used, a tolerably close agreement of the expansion curve with the theoretical may be expected; while a *wide* divergence, even with an early cut-off, should be viewed with suspicion.

Leakage.—We have seen that under the conditions which ordinarily obtain, cylinder condensation and re-evaporation will cause the latter part of the expansion curve to rise above that due to the isothermal expansion of the volume of steam in the cylinder at cut-off. Hence, when the expansion curve falls *below* the theoretical, as in Fig. 20, it is strong presumptive evidence of leakage either past the piston, or through the exhaust valve when separate admission and exhaust valves are used. But unless the amount of leakage is large, it is not possible to readily detect its existence in this way; also it is quite possible for leakage past the steam valve to so compensate for the leakage past the piston as to result in the production of an expansion curve in fairly close agreement with the theoretical. However, this latter coincidence is

FIG. 20.

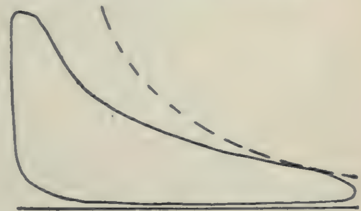

FIG. 21.

exceedingly unlikely to occur at both ends of the cylinder, and if the indications from either end suggest leakage, both piston and valves should be tested for steam-tightness at the earliest opportunity.

Leakage past the steam valve is difficult to discover from an inspection of the expansion curve, since the resulting effect—an elevation of pressure near the end of the expansion period—is analogous to that produced by re-evaporation. With a deviation as decided as that shown in Fig. 21, leakage past the steam valve would be at once inferred, even under conditions most favourable to re-evaporation. A much less marked departure from the theoretical would be permissible if the piston speed was fairly high—500 or 600ft. per minute,—as under such conditions less time is available for re-evaporation. Indeed, with all but small engines, in diagrams taken at this speed of piston, with a cut-off at

about $\frac{1}{4}$ stroke and the cylinder supplied with dry steam, the terminal pressure should not exceed the theoretical terminal by more than 4 or 5lb., and this amount should be still less if the cylinder is efficiently jacketed.

For many purposes the application of the following test will save the further labour incurred in drawing the complete hyperbolic expansion curve:—Select a point on the expansion curve as R, Fig. 22, well in advance of the point of release, and another point on the curve as Q, decidedly after the point of cut-off. From R draw a vertical line meeting a horizontal from Q in the point S. Join S to Z, the point of intersection of the clearance line with that of zero pressure. From R draw a horizontal cutting S Z in *n*,

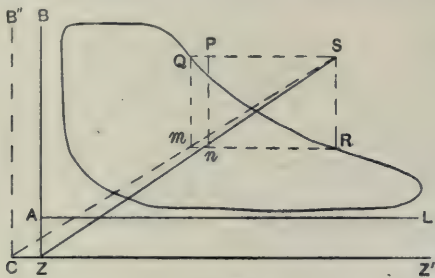

FIG. 22.

and from *n* draw a perpendicular *n* P. From the previous construction (Fig. 18) it will be seen that P is a point in the standard curve drawn through R, and hence, from what has been said, the extent to which P fails to coincide with Q will afford an indication of the probability or otherwise of the occurrence of leakage, valve leakage and re-evaporation causing P to lie to the right, and piston leakage to the left, of Q.

Clearance.—It will have been noted that for the construction of the theoretical curve the amount of the clearance volume of the cylinder is required. This is frequently unknown, and failing the opportunity for measuring it either from the actual cylinder or the drawings, it will be necessary to assume a reasonable value according to the size and type of cylinder.

The ordinary limits of cylinder clearance volumes for various types of engines are as follow:—

Type of Engine.	Clearance Per Cent.	Type of Engine.	Clearance Per Cent.
Corliss valves . . .	2 to 4	Locomotives	7 to 12
Long slide valves . .	$1\frac{3}{4}$ to $2\frac{3}{4}$	Small horizontal and vertical	9 to 20
Double beat valves . .	5 to 7	Flat slide valves (marine) .	10 to 18
Ordinary slide valves .	6 to 12	Piston valves	12 to 25

A method is sometimes advocated for finding the position of the clearance line from the expansion curve. Thus, taking the two points R and Q in Fig. 22, and drawing horizontals and verticals from each, the rectangle Q S R m is obtained. The line joining S and m, when produced, cuts Z Z′ in C, and C B″ is then assumed to represent the clearance line. It is clear from the basis of such a construction that

FIG. 27.

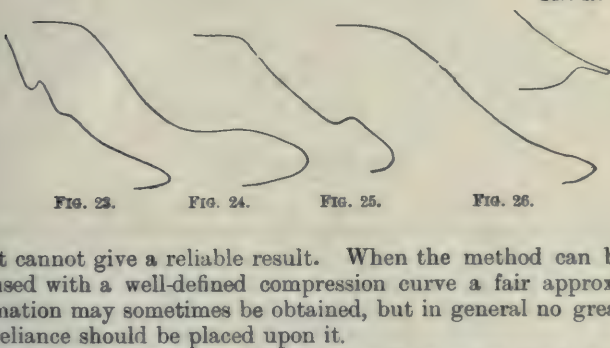

FIG. 23. FIG. 24. FIG. 25. FIG. 26.

it cannot give a reliable result. When the method can be used with a well-defined compression curve a fair approximation may sometimes be obtained, but in general no great reliance should be placed upon it.

Distorted Expansion Lines.—It will be noted that with cylinder condensation, re-evaporation, piston leakage, and valve leakage, the separate or combined effect is generally manifested by a *gradual* change in the direction of the curve. Abrupt breaks in the expansion line are, however, met with, and these are usually due to defects in the valve gear. A local distortion similar to that shown in Fig. 23, which is sometimes found in diagrams from cylinders provided with separate steam and exhaust valves, is usually due to the steam valve reopening after the nominal cut-off, the valve not being effectually closed, owing to defects in the valve or seating, until late in the stroke. In engines fitted with

back expansion valves, more or less abrupt changes in the expansion curve, but later in the stroke, as in Figs. 24 and 25, are frequently due to the expansion slide reopening the port in the main slide before the latter closes the port in the cylinder. Similar effects are produced in diagrams from engines fitted with the Rider expansion gear, by reason of the expansion slide rotating too far; also in engines with badly arranged valves of the gridiron type. Occasionally in engines fitted with ordinary slide valves a binding of the valve or rod at some point in the expansion period may result in a temporary lifting of the valve from its seating, and the consequent readmission is indicated as in the previous cases. Leakage during and for some time after the point of cut-off frequently results in a reversal of curvature of the expansion curve during the first part of its length (Fig. 26). In general, small irregularities will be detected in this portion of the curve unless the piston speed is fairly high. Release delayed beyond the end of the stroke will cause the curve to double back upon itself, or nearly so, resulting in a formation similar to that shown in Fig. 27.

The Release Line.

The form assumed by the release line of the diagram depends very largely upon the point of cut-off, inasmuch as this determines the terminal pressure. The piston speed employed is also, however, a not unimportant factor. With very slow moving engines, expansion may be continued practically to the end of the stroke, in which case the release line may be almost vertical, as in Fig. 28. In most cases, however, the discharge of the exhaust steam occupies sufficient time to render it desirable to open communication with the exhaust port before the end of the stroke is reached. This has the effect of materially lowering the back pressure, since the cylinder is more completely cleared of the exhaust steam before the return stroke commences. Thus, in Fig. 29 the broken lines may be taken to indicate the form of the release end of the diagram when expansion is continued to the end of the stroke, while the full lines show the result of opening to exhaust at the point *a*. In the latter case the rapid fall of pressure results in the formation of the release

line $a\,c$, the pressure falling during the return stroke to the point e in the exhaust or back pressure line. It will be seen that so far as the area of the diagram (and hence the work done in the complete stroke) is concerned, the gain of the area $d\,c\,e$ far more than offsets the loss of the area $a\,b\,d$.

Pre-release.—The amount of "lead to exhaust," or "pre-release," in Fig. 29 is represented by the horizontal distance f. It is frequently expressed as a fraction of the working stroke, and varies in amount from 2 to 25 per cent. It will be evident that as the piston speed is higher the exhaust lead should be increased, since the time available for the removal of the steam is correspondingly reduced. Similarly, as the difference becomes greater between the virtual final pressure (as $g\,b$, Fig. 29) and the back pressure, the opening to exhaust should take place earlier in the stroke. This

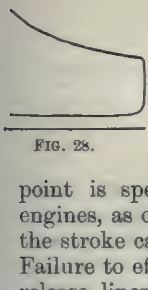

FIG. 28.

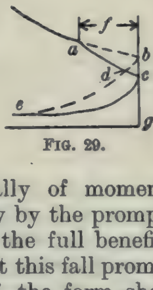

FIG. 29.

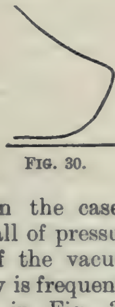

FIG. 30.

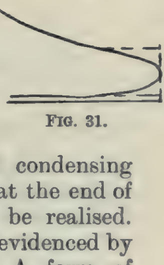

FIG. 31.

point is specially of moment in the case of condensing engines, as only by the prompt fall of pressure at the end of the stroke can the full benefit of the vacuum be realised. Failure to effect this fall promptly is frequently evidenced by release lines of the form shown in Fig. 30. A form of release line frequently met with is shown in Fig. 31, the exhaust opening occurring at such a point as will allow about one-half the fall of pressure to take place during the forward stroke, and the remainder during the early part of the return stroke. In slide valve engines, however, this type of release line is usually the outcome of an endeavour to compromise between the addition of lap in order to secure adequate compression, and the removal of lap to give an early release. When separate admission and exhaust valves are used, the problem of reconciling these conflicting factors is simplified considerably.

Instances of release occurring too early in the stroke are comparatively rare. An example is given in Fig. 32, taken

from the high-pressure cylinder of a mill engine fitted with piston valves. The dotted lines suggest the extent by which the diagram area might have been increased by a more judicious arrangement.

When the expansion is such that the terminal pressure is equal in amount to the back pressure, the release line will practically disappear, the toe of the diagram being of the form shown in Fig. 33. If the cut-off takes place still earlier in the stroke, the expansion curve crosses the back pressure line as shown in Fig. 34. At the point of exhaust opening the pressure in the cylinder is less than that due to the back pressure, and hence the release line (which in this case may be regarded as *negative*) is drawn in an upward direction, indicating a rise instead of a fall, to the back pressure line. The form assumed by the release line

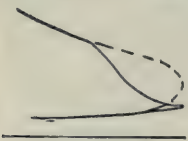

Fig. 32.

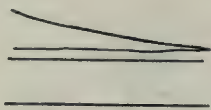

Fig. 33.

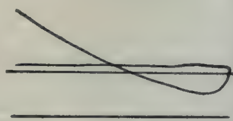

Fig. 34.

when the opening to exhaust takes place *after* the end of the stroke is reached has already been indicated in Fig. 27.

Just as in the case of the steam line contracted port area gives a downwardly sloping line, indicating the tardy and impeded entry of the steam, so also will insufficient exhaust port area cause a gradual downward sloping of the release line. In particularly bad cases the obstruction to the steam flow may be sufficient to prolong the release until compression commences, as shown in Fig. 35. It is evident that this is a defect which may impair the efficiency of the engine to a greater extent than wiredrawing during admission.

The Exhaust Line.

From what has been said in the immediately preceding section, it will be seen that the back pressure opposing the motion of the piston during its return stroke will depend upon (*a*) the lead to exhaust or amount of pre-release; (*b*)

the terminal pressure; and (c) the area of the exhaust port and connected piping. It will also depend upon (d) the piston speed; (e) the efficiency of the condenser in condensing engines; and (f) the amount of compression adopted. In general, increasing (a), (c), and (e) will diminish the back pressure, while increasing (b), (d), and (f) will augment it. The influence of the last-named factor, however, requires some further consideration, as will be shown in the section on the compression line.

Under the most favourable conditions the exhaust line of the diagram from a non-condensing engine may appear to almost exactly coincide with the atmospheric line. More usually, however, the interval between the two is distinctly evident, the back pressure line representing in the best cases some 2 or 3lb. pressure per square inch above that due to

FIG. 35.

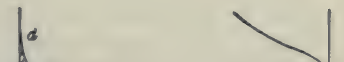

FIG. 36.

the atmosphere. As previously pointed out, late opening to exhaust, together with insufficient exhaust port area, may result in a gradual falling line of pressure during the whole of that part of the return stroke performed before compression commences (Fig. 35); but in general a portion of the lower line of the diagram will be approximately parallel to the atmospheric line, as in Fig. 36.

It has already been explained that for convenience in diagram analysis the line *a b c d* (Fig. 36), described during the return stroke, is regarded as composed of (1) the completion of the release line *a b*, (2) the exhaust line *b c*, and (3) the compression line *c d*, and that in general parlance the whole of the line *a b c d* is designated the back pressure line. From what has been said previously, it will be seen that as the line *a b c d* represents the variation of the pressure opposing the motion of the piston, the shaded area in Fig. 36 represents the back pressure work done over and

above that due to the atmospheric pressure. Hence, so far as the amount of power developed is concerned, it becomes highly desirable to reduce this area as far as practicable. With ample exhaust port area and free connections therefrom, the pressure represented by the height of *b c* above A L may often be reduced almost to the vanishing point, but the larger area under the release line *a b* cannot in general be materially reduced except at the cost of reducing the forward pressure, even if the valve gear will enable the necessary early release to be effected. Similarly, a diminution of the area under the compression curve *c d* involves the loss of a larger quantity of steam used in filling the clearance space, and consequently diminishes the work done per pound of steam used. Further, the absence of the cushioning effect due to compression may have a prejudicial effect upon the running of the engine, while incidentally, smaller losses, such as those resulting from the longer exposure of the cylinder to the atmosphere or the condenser, and the absence of the heating effect due to compression, are also involved. Thus it will be seen that the various events of the stroke are largely mutually dependent upon one another, while to some extent the influences of other factors also require to be taken into consideration.

Directing attention mainly to the horizontal portion of the back pressure line, mention should be made of the loss of power which results from improperly arranged systems of piping for heating by exhaust steam. The resistance to the steam flow offered by a needlessly large number of elbows and long and winding lengths of piping of insufficient diameter not infrequently increases the back pressure to an extent which more than nullifies the apparent gain due to exhaust heating; on the other hand, when suitable provision is made to secure an almost unimpeded discharge of the exhaust, its heating effect may often be turned to useful account at the expense of some 1 or 2lb. additional back pressure.

In condensing engines the back pressure will obviously depend upon the vapour pressure existing in the condenser —in other words, upon the excellence of the vacuum,—while this in turn depends upon the temperature at which the condenser can be advantageously maintained. The table

below, giving the absolute vapour pressures corresponding to various temperatures, will be of service in this connection.

From the table it will be seen that a condenser temperature of 102° F. corresponds to a vapour pressure of 1lb. per square inch, while, when the temperature rises to 127° F., the pressure increases to 2lb. per square inch. Thus with an increase of 25° the vapour pressure is more than doubled.

TABLE II.—VAPOUR PRESSURES FOR VARIOUS TEMPERATURES.

Temperature of Condenser.	Pressure of Vapour. Lb. per Sq. In.	Temperature of Condenser.	Pressure of Vapour. Lb. per Sq. In.	Temperature of Condenser.	Pressure of Vapour. Lb. per Sq. In.	Temperature of Condenser.	Pressure of Vapour. Lb. per Sq. In.
85° F.	0·590	97° F.	0·860	109° F.	1·229	121° F.	1·730
86	0·609	98	0·887	110	1·265	122	1·779
87	0·629	99	0·914	111	1·302	123	1·828
88	0·650	100	0·943	112	1·341	124	1·879
89	0·671	101	0·972	113	1·381	125	1·931
90	0·692	102	1·001	114	1·421	126	1·984
91	0·715	103	1·031	115	1·462	127	2·039
92	0·738	104	1·062	116	1·504	128	2·096
93	0·761	105	1·094	117	1·547	129	2·154
94	0·785	106	1·126	118	1·591	130	2·213
95	0·809	107	1·159	119	1·636	131	2·273
96	0·834	108	1·193	120	1·682	132	2·335

It is clear that the back pressure in a condensing engine can never be less than the vapour pressure in the condenser, the latter constituting what is virtually a greatly attenuated atmosphere. In estimating the amount of back pressure due to the frictional resistance offered to the exhaust steam flow, it is convenient to refer the back pressure to a "vacuum" line, as S T in Fig. 37, representing by its distance below A L the mean pressure equivalent to the "inches of vacuum" shown by the vacuum gauge attached to the condenser, and sometimes expressed as "pounds of vacuum."

For the purpose of comparing the back pressure actually realised with the lowest attainable, the addition of the "vacuum" line will suffice; but it is often necessary to draw in also the line of perfect vacuum or zero line Z Z′, and usually this is located at a distance below A L equivalent to a pressure of 14·7lb. per square inch on the scale of the diagram. The actual pressure, however, will vary from time to time, and if exactness is desired Z Z′ must be drawn so that the pressure Z A is equivalent to the existing barometric pressure. The pressure Z S is then that due to the vapour in the condenser.

Air Leakage.—In indicating condensing engines, the back pressure shown in the diagram from the front end of the cylinder will often be found to be materially higher than that shown in the card taken from the back end. This in general is due to leakage of air which finds its way past

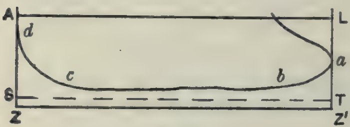

FIG. 37.

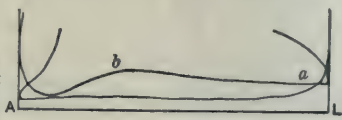

FIG. 38.

the gland of the piston rod into the cylinder, and thus impairs the vacuum at that end. A rod-packing which effectually prevents the escape of steam from the cylinder is not necessarily equally efficient in preventing the ingress of air during the return stroke, although this is an assumption commonly made. Air leakage past indicator cocks, drain cocks, and snifting valves may also occur. The importance of preventing any leakage of air into the condenser is too obvious to need insisting upon; cases, however, are sometimes met in which even ordinary precautions are neglected, and the air-pump is incapable of effecting the removal of the increased volume of air. The design of the air-pump has also an important effect upon the maintenance of the vacuum. The clearance should be as small as possible, and in horizontal pumps care must be taken to place the foot valves at the lowest possible part of the water chamber, and the delivery valves in the highest part of

the pump chamber. All pockets in which air might collect should also be scrupulously avoided.

Typical Exhaust Lines.—Generally the actual configuration of exhaust lines met with under normal conditions will approximate to the lines shown in Figs. 36 and 37. Occasionally, however, the outline will assume the form indicated at *a b*, Fig. 38, remaining for some time almost parallel to the atmospheric line, and afterwards falling somewhat before the point of exhaust closure is reached. If, as is usually the case, the diagram from the other end shows an ordinary type of exhaust line, it is fair to conclude that the discrepancy in the first case is due to the unequal travel of the valve, the opening to exhaust being insufficient at the one end, and more than is required at the other. Unequal degrees of cut-off would, however, also appear in this case,

FIG. 39.

FIG. 40.

unless the valve had unequal outside laps. Exhaust lines of the form shown in Fig. 39, which are not infrequently found, may usually be taken as indicating a throttling of the exhaust, the effect becoming intensified as the slide valve reaches the limit of its movement in the direction of opening. Over-travel of the valve would produce this effect, owing to the valve face covering the exhaust port to such an extent as to greatly reduce the effective opening of the latter. Too narrow an exhaust port will be somewhat similarly indicated. The addition of inside lap in order to give compression will also lead to throttling of the exhaust, and this cause would therefore be suspected if the compression period was well defined. It will be noted that in each of these instances the effect is accentuated by the higher speed of the piston near mid-stroke coinciding approximately with the extreme limit of the valve movement, thus rendering the throttling more pronounced at the position

named, and also intensifying it at higher piston speeds. In engines fitted with Corliss valves a similar action may occur owing to the angular movement of the valve being too great.

A somewhat similar rise in the centre of the exhaust line, although generally to a less marked extent, is sometimes met with in cases where the same exhaust pipe serves for two cylinders working with cranks at right angles. Here the sudden release of steam from the one cylinder taking place when the other is about at mid-stroke, causes a momentary increase of back pressure, which produces the rise referred to. The effect is sometimes seen in diagrams from locomotives, more particularly when cut-off occurs late in the stroke, and the release pressure is consequently high. With slide valves having different amounts of inside lap the resulting exhaust lines will differ correspondingly (Fig. 40), the release in the one case being delayed and the compression very pronounced; while in the opposite diagram release occurs early and compression is small, the other events of both strokes taking place in correct time. This, however, is a defect but seldom met with.

The Compression Line.

Although it is practically impossible to reduce the percentage of cylinder clearance below certain limits, the loss of economy from this cause may be diminished very considerably by closing the exhaust valve before the end of the stroke is reached. By this means the volume of steam enclosed in the cylinder is compressed into the clearance space as the piston completes its stroke, its pressure rising in consequence from that of the back pressure at the point of exhaust closure to some percentage of the initial pressure, the precise amount depending mainly upon the exhaust or back pressure, the percentage of clearance, and the fraction of the stroke performed under compression. The compressed steam increasing in pressure as its volume diminishes, offers a rapidly increasing resistance to the motion of the piston, and in this way assists in bringing the reciprocating masses quietly to rest at the end of the stroke. The value of cushioning as an effect conducing to smooth running is very considerable, especially in high-speed engines. In slow-

moving engines, however, only a moderate amount of compression is usually allowed.

Viewed as a factor in the economical use of steam, the amount of compression which will yield the highest efficiency is not readily determinable. Experiments carried out by Professor Dwelshauvers-Dery with saturated steam appear to show that compression, so far from being of value, involves a loss in a direct ratio to the degree of compression employed. With superheated steam, however, the conclusions reached were, that if the indicated work per stroke was maintained constant (by suitably varying the steam pressure), the degree of compression is without influence upon the steam consumed per I.H.P. These conclusions, which

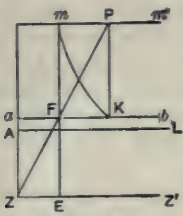

FIG. 41.

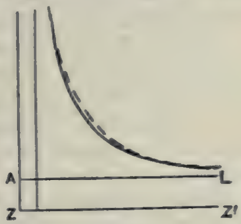

FIG. 42.

exhibit such a wide divergence from the views generally held, require ratification by other and more exhaustive tests before they can be fully accepted ; but there is no doubt that the discrepancy between theory and practice is, in this case, attributable to interchanges of heat between the steam and the cylinder walls. General evidence of this is afforded by the better results obtained with superheated than with saturated steam, indicating a nearer approximation to the theoretical, as the steam more nearly approaches the condition of a perfect gas. Hence, in the absence of special knowledge upon this point, compression should be regulated mainly with regard to the degree of cushioning required to secure smooth running of the engine.

Assuming compression to follow the hyperbolic law (page 12), the following simple method is available by which to determine the point of exhaust closure as K (Fig. 41) for any required final compression pressure as E m :—Draw

a line from Z passing through F, the point in which the final pressure ordinate E m intersects the line of back pressure $a\,b$. Draw a horizontal $m\,m'$ at a height representing the required compression pressure E m, and meeting Z F produced, in P. From P drop a perpendicular meeting the back pressure line in K, which is then the required point of exhaust closure. In practice, however, it will generally be found that the compression pressure falls somewhat short of that given by this method.

With steam fairly dry, and other conditions favourable, the compression curve when well defined will often show a close approximation to the hyperbolic curve. This is particularly evident in diagrams from locomotives when linked up, the ordinary valve gear giving increased compression as the cut-off takes place earlier in the stroke. Fig. 42 shows the compression curve of a diagram taken from a goods engine with cylinders 16 by 22in., running at a piston speed of about 900ft. per minute. The close agreement with the theoretical compression curve (shown dotted), which is very evident, is quite typical of diagrams from locomotives.

An excessive amount of compression usually results in the formation of a loop as shown at *d* and *e*, Fig. 8, or a peak is formed as *f*, Fig. 8, the pressure rising above that in the steamchest. When the valve opens to steam, the pressure falls as shown, owing to the momentary escape of the compressed steam, and the consequent fall of pressure to that of the incoming supply. A total absence of compression, giving a perfectly square corner, is rarely met with, sufficient steam being usually confined to give a well-rounded corner, as *a*, Fig. 8.

More or less sudden deviations from the compression curve, such as indicated in Fig. 43 (*a* to *e*), are ascribable to condensation of the cushion steam, or to leakage past the piston or past the valve in engines provided with separate exhaust valves. Of course, all these causes may co-exist, and it is somewhat difficult to estimate their relative effects upon the shape of the compression line. In general, however, such a small drop as that shown in *a* may be taken as due to condensation owing to the diminished volume of steam giving up heat to the now relatively much larger

clearance surfaces. This effect, coupled with late admission, results in the formation shown in *b*, but it should be noted that a very similar effect is produced when the piston covers the opening to the indicator at the end of the stroke. (See Part I. of this work, page 80.) Such pronounced deviations as shown at *c*, *d*, and *e*, Fig. 43, however, are probably due to leakage, and if a comparison of the expansion curve with the theoretical confirms this view, the piston should be secured in various positions in the cylinder by blocking the crosshead or crank, when leakage can be detected by removing one of the cylinder covers and allowing steam to enter the opposite end of the cylinder. Occasionally the piston may appear fairly steam-tight in some parts of the stroke,

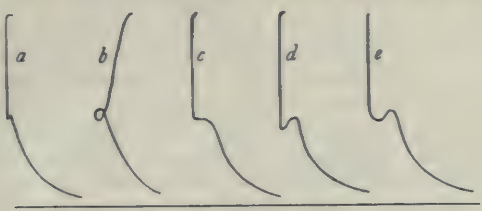

FIG. 43.

while leaking badly in others; but, as a rule, if marked leakage occurs in any one position, it occurs to a greater or less extent in adjacent positions. Hence, when such lines as those at *c* and *e* are met with, the early part of the compression curve usually falls well below the hyperbolic curve, drawn through the point at which compression commences.

The compression curve usually approaches more nearly to the theoretical in high-speed engines than in low, since there is less time available for the transference of heat from the steam to the cylinder walls, and from the latter to the condenser or atmosphere, as the case may be. Closer approximation to the theoretical is also found in jacketed than in unjacketed cylinders, and in general any influence which improves the quality of the steam will have a similar effect upon the compression curve.

CHAPTER III.

DIAGRAM ANALYSIS.

IN the preceding chapter the various periods of the indicator diagram have been independently considered. It is necessary, however, to keep in view the fact that under all ordinary conditions of practice the several events occurring during the complete double stroke of the piston are more or less mutually dependent upon one another. More particularly is this the case when the steam distribution is effected by the simple slide valve gear so extensively employed, the basis principle of which may be said to be common to the majority of other valve gears. Hence it will be convenient at this point to consider the effects produced by variously modifying the valve gear, and for this purpose the tabulated statement given on page 49 will be found of service.

The four variations of which the effects are *separately* considered are: (1) Increase of outside lap, (2) increase of inside lap, (3) increase of the angular advance of the eccentric, and (4) increase of the valve travel. It will be seen that 1 and 2 refer to modifications of the dimensions of the *valve*, and 3 and 4 to modifications of the dimensions or position of the *eccentric*.

It will be noted that the effect of increasing the *angular advance* is to cause all the events to occur earlier in the stroke, but the respective periods are unchanged. A similar effect is produced by increasing the *lead* without altering the outside lap.

If the lead is to be increased, while both valve travel and cut-off are to be unaltered, the lap must be reduced by *one-half* the desired increase of lead.

By Increasing the	Admission	Expansion	Exhaust	Compression
Outside Lap.	Commences later. Ceases earlier. Period reduced.	Commences earlier. Ceases as before. Period increased	Unchanged.	Commences as before. Period depends on admission point.
Inside Lap.	Unchanged.	Commences as before. Ceases later. Period increased.	Commences later. Ceases earlier. Period reduced.	Commences earlier. Ceases as before. Period increased.
Angular Advance.	Commences earlier. Ceases earlier. Period unchanged.	Commences earlier. Ceases earlier. Period unchanged.	Commences earlier. Ceases earlier. Period unchanged.	Commences earlier. Ceases earlier. Period unchanged.
Valve Travel.	Commences earlier. Ceases later. Period increased.	Commences later. Ceases earlier. Period reduced.	Commences earlier. Ceases later. Period increased.	Commences later. Ceases earlier. Period reduced.

As may be readily seen from the above tabular statement, the effect of giving inside clearance (sometimes designated "negative inside lap") to a slide valve is to prolong the exhaust period, with a corresponding diminution (equally shared) of the expansion and compression periods.

The foregoing remarks (which with obvious modifications are applicable to diagrams other than those from simple engines) will be found of service in determining the causes of many of the malformations of the diagrams now to be considered.

Defective Valve Setting.

Plain Slide Valve Gear.—Two types of defects are frequently met with in indicator diagrams from slide valve engines which are traceable to defective valve setting: (*a*) Those due to the incorrect placing of the valve on the spindle, and (*b*) those due to the incorrect placing of the eccentric upon the crankshaft. Diagrams exhibiting a combination of these typical defects are, however, still more

common than those displaying only one or other of the phases of defective valve setting here considered.

A brief consideration will show that the conditions first named will involve an unequal distribution of steam to the two ends of the cylinder, since at one end of the valve the outside lap will be too great, and at the other too small. Hence in the first case steam will be admitted too late and cut off too early, while in the other end of the cylinder steam will be admitted too early and cut off too late. The resulting diagrams will therefore be similar in character to those shown in Fig. 44, and from which will also be seen the effect produced upon the exhaust side. It is evident that any increase of outside lap due to the displacement of the valve on its spindle will entail a corresponding diminution

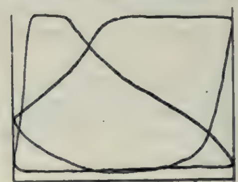

FIG. 44.

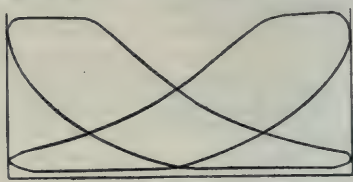

FIG. 45.

of the exhaust lap at that end of the valve. Hence in the diagram showing late admission and early cut-off, above referred to, the exhaust will be found to commence early and cease late, giving little or no compression. The diagram from the opposite end will show faults of an opposite character — late opening and early closing to exhaust. From this it will be seen that when the valve is incorrectly fixed on the valve spindle the defects shown by the diagrams from *opposite* ends of the cylinder will be *opposite* in character.

The case has now to be considered in which the valve is correctly set so far as its position on the spindle is concerned, but where the eccentric is wrongly secured upon the shaft. Reference to the tabular statement given on page 49 will show that if the eccentric is in advance of its true position —*i.e.* if the angular advance is too great,—*all* the events of the stroke on *both sides* of the piston will take place too early. Conversely, if the angular advance is insufficient, all

the events on *both* sides of the piston will be delayed. Hence, in the first case the diagrams will partake of the form shown in Fig. 45, while in the second case the form will approximate more to those shown in Fig. 46. It is clear that when reasonable limits are not exceeded the modification of the steam distribution shown in Fig. 45 is not to be regarded as disadvantageous. The same cannot

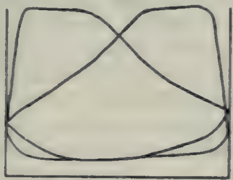

Fig. 46.

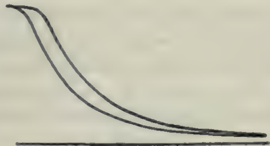

Fig. 47.

be said, however, in regard to the cards shown in Fig. 46, for here tardy admission and release results in a very material reduction of the diagram area. Extreme cases of excessive and deficient angular advance are given in Figs. 47 and 48 respectively. In the last-named figure steam is admitted at a, but having to follow a now rapidly moving piston the full pressure is not attained until near the end of the stroke. Cut-off occurs at b, the expansion line extending from b to c, and afterwards doubling back nearly upon itself until d, the point of release, is reached. The pressure falling to e, the exhaust line $e\ f\ g$ is formed. At g, communication with the exhaust ceases, and the pressure falls behind the advancing piston until the point a is reached and steam is again admitted.

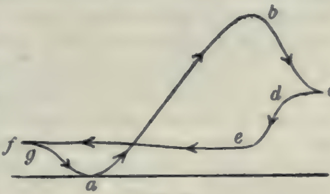

Fig. 48.

A consideration of Figs. 44 and 46 will enable a general distinction to be drawn between diagrams showing incorrect location of the valve on the rod, and those indicating a misplacement of the eccentric on the crankshaft. In the first case the admission lines are roughly parallel to each other; in the second case, these lines slope in opposite directions.

It is important to distinguish between the effects due to the displacement of the valve on its spindle on the one hand, and to unequal amounts of lap on the other. In the first case, as a reference to Fig. 44 will show, there is a virtual decrease of outside lap at one end, causing a prolonged admission, but it is accompanied by a very late release and an early compression. In the diagram from the opposite end of the cylinder, the conditions are entirely reversed, the shorter admission period being followed by an early release, and little or no compression. With a valve constructionally provided with unequal laps, however, the conditions as to admission and cut-off may closely resemble the foregoing, but the point of exhaust opening and closing will be approximately similar at each end. Cases in which unequal inside lap is provided are more rare. Under these circumstances, the diagrams, while approximately balanced so far as the points of admission and cut-off are concerned, will show decided differences in regard to the points of release and compression.

In this connection it should be observed that by reason of the angular vibration of the connecting rod, exact equality in the steam distribution will not ordinarily be obtained in both ends of the cylinder of a direct-acting slide-valve engine. The amount of the inequality will increase as the ratio of the lengths of the connecting rod and crank decreases, the effect being most marked in the cut-off, which is delayed in the forward or outward stroke, and accelerated in the return or inward stroke. This inequality may be rectified by reducing the lap on the outer or front end of the valve, but it will be noted that this in turn destroys the equality of lead, the amount of pre-admission in the return stroke being increased correspondingly with the reduction of the lap. But in many cases approximate equality of cut-off is of more importance than equality of lead, while in the usual type of marine or inverted-cylinder engine the lap is often made still more unequal, so that the excess of effort exerted on the underside of the piston assists to balance the effect of the weight of the reciprocating parts of the engine. Frequently, in large marine engines, the valve, which is made symmetrical, is moved slightly upwards along its rod. This not only gives

the required difference in the leads, but also increases the inside lap at the bottom end of the valve, thus increasing the compression period, and giving greater cushioning effect to the descending weights.

It is also worthy of note that in this type of engine the effects of wear of the eccentrics, links, and bearings, all tend to cause the valve to move in a downward direction, and thus to diminish the lead at the lower end and to increase it at the upper end of the cylinder. In horizontal engines the effect of wear will be in the same direction, but in a less marked degree. In a badly-worn link motion, however, "play," or "lost motion," will delay the opening to steam at each end, thus virtually reducing the lead.

The exhaust side of the valve can be similarly modified by adding lap or giving inside clearance, and in this way

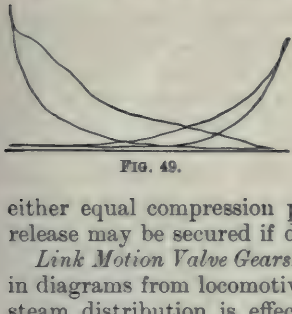

FIG. 49.

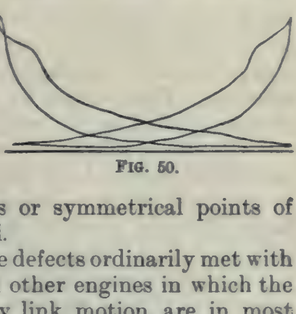

FIG. 50.

either equal compression periods or symmetrical points of release may be secured if desired.

Link Motion Valve Gears.—The defects ordinarily met with in diagrams from locomotive and other engines in which the steam distribution is effected by link motion, are in most cases somewhat analogous to those already discussed. The effect of any displacement of the valve on its spindle is, however, more marked, especially when the cut-off occurs early in the stroke. A case in point is shown by the diagrams Figs. 49 and 50, which were taken from a locomotive with 18 by 24in. cylinders, running at 200 revolutions per minute. As will be seen, there is a considerable difference in the areas of the two diagrams shown in Fig. 49, which, however, is entirely absent in the cards taken after adjusting the position of the valve on the spindle (Fig. 50).

The somewhat flexible character of the link motion, and the number of joints involved in its construction, tend to the accumulation of lost motion; further, the slight though

definite alterations in the relative positions of the centres of the gear and of the cylinder and driving axle (more marked when the cylinders are inclined) under variations of load and steam pressure, also introduce irregularities in the steam distribution, although in general they are of comparatively small moment. A much more marked effect results from

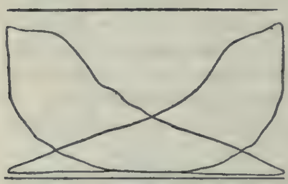

FIG. 51.

FIG. 52.

the deficient lubrication of the slide valve, causing springing of the rods (especially when these have a "set-off," as in many American engines), and often seriously affecting the diagram. Figs. 51 and 52 afford a good illustration of the case in point, Fig. 51 being a card taken when the valve was effectually lubricated, while Fig. 52 shows the marked alteration produced by an insufficient supply of lubricant. It may be added that these diagrams show that the average mean effective pressure is reduced from 70lb. per square inch in Fig. 51 to 51·8lb. per square inch in Fig. 52.

At high speeds the diagram is often modified very considerably, the position of the point of cut-off being

FIG. 53.

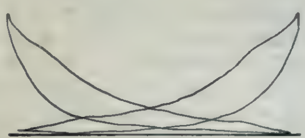

FIG. 54.

apparently changed, and the other events of the stroke altered in a more or less marked degree. This is seen very clearly in Figs. 53 and 54, taken from a locomotive having cylinders 17 by 24in., and running at speeds of 15 and 45 miles per hour, all other conditions remaining unchanged.

The cut-off, which is nominally at $\frac{1}{4}$ stroke, is fairly well defined in Fig. 53. In Fig. 54, however, the cut-off occurs much earlier, the event apparently taking place at about $\frac{1}{7}$th of the stroke. Among other modifications noticeable in the diagrams is the marked increase in the wiredrawing as the speed increases, and also the accompanying increase in the degree of compression. The combined effect upon the area of the diagram is such that the average mean effective pressure is reduced from 44·8lb. per square inch in Fig. 53 to 23·5lb. per square inch in Fig. 54.

These characteristics become still more marked when, as is more usual in practice, higher speeds are accompanied by earlier cut-offs. The effect produced upon the steam distribution by "linking-up" is represented by a shortening of the valve travel and a simultaneous increase in the angle of

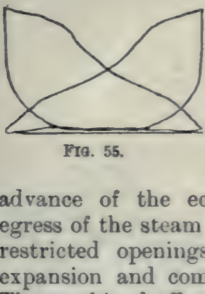

FIG. 55.

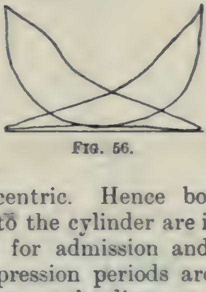

FIG. 56.

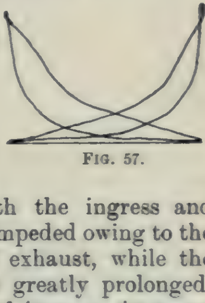

FIG. 57.

advance of the eccentric. Hence both the ingress and egress of the steam to the cylinder are impeded owing to the restricted openings for admission and exhaust, while the expansion and compression periods are greatly prolonged. The combined effect upon the diagram of increase in speed and earlier cut-off is well shown in Figs. 55 to 57, taken from an express locomotive.

One effect of giving valves inside clearance is often noticeable in the compression curves of diagrams taken at low speeds and with the gear in the full forward or backward notch. This is indicated in Fig. 58, the change in the direction of the compression curve shown at B being due to the steam from the opposite end of the cylinder "blowing over" during the brief interval in which the two ends of the cylinder are in communication. The effect, however, is only observable in diagrams taken at comparatively low speeds.

Another "blowing-over" effect seen in low-speed cards

from locomotives is the hump or rise in the back-pressure lines, an example of which is afforded by Fig. 59, taken from an engine with cylinders 18 by 24in. (G.E.R.) running at four miles per hour. In this case the temporary rise in back pressure is due to the release from the opposite cylinder momentarily increasing the pressure in the exhaust pipe. This distortion becomes less marked as the terminal pressure is reduced either by linking up or by the throttling due to increased speed, and hence is only noticeable at low speeds and late cut-offs. If the speed remains uniform and the cut-off is shortened, the consequent earlier release from the opposite cylinder causes the hump gradually to approach the exhaust end of the diagram.

The resistance offered to the exhaust by the blast pipe nozzle is also among the important influences affecting the

FIG. 58.

FIG. 59.

economical use of steam in the locomotive, but its effect upon the diagram is not greatly dissimilar from that already discussed in Chapter II.

Expansion Valve Gear.—Engines fitted with a separate expansion or cut-off valve working on the back of a main distributing valve, as in the Meyer gear, not infrequently furnished cards showing a distorted expansion line at the toe of the diagram, as in Fig. 60. This effect is produced by the expansion valve overrunning the opening in the main valve before the latter closes communication with the steam port, thus allowing a readmission of steam to the cylinder as indicated. Insufficient outside lap on the main valve (resulting in a late opening to exhaust. and little or no compression) is one cause of this defective action; but even with a sufficiency of lap it may also occur by reason of the cut-off plates being too narrow. This fault is also likely to be met with in diagrams from engines

in which the stroke of the expansion valve is varied by an alteration of the throw of the expansion eccentric, as in the Hartnell type of valve gear.

A somewhat similar distortion of the expansion curve is obtained when the main valve is displaced from its correct position on the spindle, causing the valve to overrun the valve face at one end (Fig. 24). This case, however, may be distinguished by the great dissimilarity of the cards from opposite ends of the cylinder.

Corliss Valve Gear.—Diagrams from engines fitted with Corliss and similar types of detachable gear should, and frequently do, show a closer approach to what may be called the ideal steam distribution than any other system of valve gear. With the gear correctly adjusted—and premising dry steam and adequate steam and exhaust port areas,—

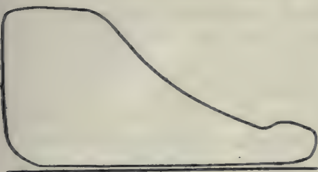

FIG. 60.

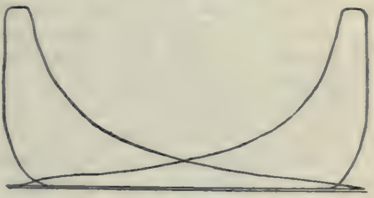

FIG. 61.

such a diagram as that shown in Fig. 61, which may be considered practically perfect, may often be obtained under favourable conditions, a result due not only to the particular method employed for operating the valves, but also to the facility for the independent adjustment of the steam and exhaust functions which this gear offers. Under these conditions it becomes possible to secure a prompt attainment of initial pressure, a well-sustained horizontal steam line which is terminated by a sharp cut-off giving an expansion curve entirely free from wiredrawing, the prompt opening to exhaust near the end of the stroke, with a rapid fall to the back pressure, and finally a suitable amount of compression to ensure smooth running.

With these facilities for separately adjusting the various events of the stroke, a faulty steam distribution is inexcusable. Nevertheless such cases are not infrequently

met with, and as an illustration of faulty valve setting and its rectification in a Corliss engine, the diagrams shown in Figs. 62 to 66 are submitted. Examining the cards from the engine as found (Fig. 62), it will be seen that in the left-hand diagram the valve does not open to steam until the crank is well past the dead centre. As this is the outward

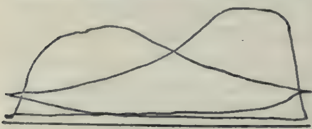

FIG. 62.

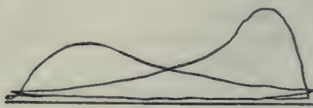

FIG. 63.

stroke, the rapidly increasing piston velocity in conjunction with the late admission results in the formation of the curved steam line shown, the highest pressure attained being only 62lb. per square inch as compared with a boiler pressure of 90lb. per square inch. The terminal pressure is high, and the release does not occur until the piston is on its return stroke. A somewhat similar condition of affairs is found in the right-hand diagram, but the admission is not quite so late, and this, together with the slower movement of the piston during the first part of the return stroke, enables a pressure of 75lb. per square inch to be obtained with an outline of a more distinctive character. The release, however, is late, and the back pressure line does

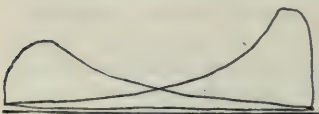

FIG. 64.

FIG. 65.

not reach its lowest limit until the end of the stroke. A card taken when the greater part of the load had been thrown off (Fig. 63) shows essentially similar features, and suggests that the eccentric had slipped back on the shaft. Fig. 64 shows the effect produced by moving the eccentric forward $1\frac{1}{2}$in., and indicates that the interpretation of the original cards was correct. The result of moving the

eccentric $1\frac{1}{4}$in. still farther forward is shown in Fig. 65, and as the right-hand card is now satisfactory, no further movement of the eccentric is required. The left-hand card still shows late admission, but by shortening the valve connection at this end, the requisite earlier opening was obtained, Fig. 66 showing the final cards taken under full load.

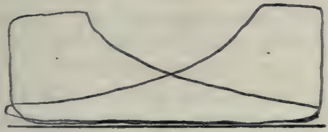

FIG. 66.

FIG. 67.

The employment of separate steam and exhaust valves in the Corliss type of engine occasionally results in the production of somewhat curious cards. In the diagram shown in Fig. 67, the failure to attain the full initial pressure during the early part of the stroke was due to the exhaust port being uncovered by its valve during the first few inches of the piston's travel. When the connection is broken between one of the exhaust valves and the wristplate, or its equivalent, the steam side of the card may be quite satisfactory, as indicated in Fig. 68; but as the steam is unable

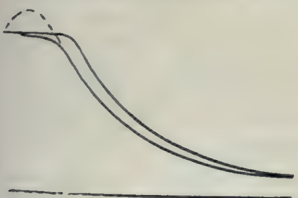

FIG. 68.

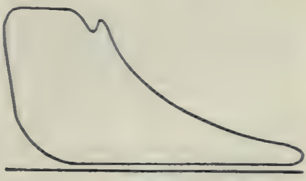

FIG. 69.

to leave this end of the cylinder, compression takes place during the whole of the return stroke. If the valves and piston are tight, the difference between the two curves will not be great, and the final compression pressure may be equal to or somewhat exceed the initial pressure. More usually, however, the loss by leakage and condensation is

sufficient to restrict the rise of pressure to within the limits indicated by the full lines.

A sluggish dashpot action will often cause a decided amount of wiredrawing at the point of cut-off, this being indicated by the rounded junction of the steam and expansion lines. On the other hand, a too energetic dashpot action will sometimes cause a rebounding effect sufficient to reopen the valve for a brief interval, resulting in the formation of a steam line such as is shown in Fig. 69.

Concluding Notes on Valve Setting.—Innumerable examples of faulty valve setting and its rectification might be given, but no useful purpose would be served thereby, as the illustrations already submitted will fully serve to indicate in a general way the method of procedure adopted in setting valves by the aid of the indicator. There are, however, a few points in this connection to which it is desirable to direct attention.

In adjusting the valve gear of slide valve engines it is important to distinguish between the effects upon the steam distribution of (1) changes in the angular advance of the eccentric, and (2) changes in the length of the valve spindle or its equivalent. These effects have already been considered, but they may be recapitulated here with advantage. The effect of alteration (1) is to cause *all* the several points in both cards to move round the diagram, so to speak, and in the *opposite* direction to that in which the eccentric is moved. The effect of alteration (2) is equivalent to increasing (or decreasing) the outside lap, while correspondingly decreasing (or increasing) the inside lap of the valve. The resultant effect of *advancing* the valve is therefore to cause cut-off and release to occur later, and compression and admission to occur earlier, in the stroke, as far as the *forward* stroke is concerned, and to cause cut-off and release to take place earlier, and compression and admission later, in the return stroke.

It is to be noted that owing to the varying velocity of the piston, either of the above alterations has a much greater influence upon points near the centre of the length of the diagram than upon events which occur near the ends of the stroke. This is a point which can be very conveniently examined by the aid of a simple valve diagram, the use of

which is also advocated for the purpose of ascertaining the resultant effect of alterations in the valve dimensions or arrangement.

It frequently happens that an alteration in the angular advance of the eccentric involves a further adjustment of the position of the valve on its spindle. This is particularly the case in alterations in the amount of lead. Thus, if of a pair of cards the lead at the crank end is excessive, while that at the opposite or head end is satisfactory, the effect of shortening the rod somewhat will be to make both leads too great, and to correct this the eccentric must be moved back so as to restore the originally correct lead to the head-end card. Assuming, for example, that the lead at the crank end is $\frac{1}{4}$in., and that at the head end $\frac{1}{16}$in., and that it is required

Fig. 70.

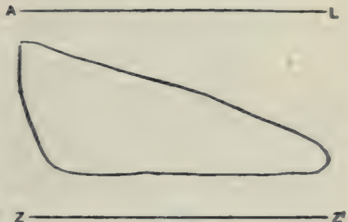

Fig. 71.

to reduce the former to $\frac{1}{8}$in., without altering the latter, this would be effected by shortening the rod $\frac{1}{16}$in., and moving the eccentric back on the shaft sufficient to move the valve $\frac{1}{16}$in.

It will be seen that the only conditions to be met by *one* alteration are (1) when both leads require equal change (effected by moving the eccentric alone); and (2) when the sum of the altered leads is to equal the sum of existing leads (effected by moving the valve on its spindle alone).

After adjusting the position of the valve on its spindle, care should be taken to firmly screw home the nuts which secure the valve in position. Carelessness in this respect may provide sufficient play or lost motion to greatly distort the card, as in the case shown in Fig. 70, which was taken from a marine engine while the nuts on the valve spindle

were slack. After tightening these, the diagram assumed the normal form as shown in Fig. 71.

Crossed or Eccentric Diagrams.—In consequence of the comparatively slow movement of the piston near the ends of the stroke, events there occurring are not shown, by the ordinary diagram, with sufficient distinctness to enable a close examination to be made of the compression and lead lines. In such a case more direct information can be derived from what are called *crossed* or *eccentric cards*, which are obtained by attaching the indicator cord to the eccentric rod, or, in the case of marine and similar engines, with cranks at right angles, to the reducing lever of the next engine. As the piston nears the ends of its stroke, the eccentric will be near the middle of its stroke and moving almost at its highest

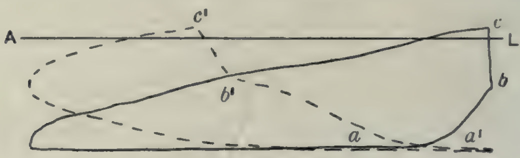

FIG. 72.

speed. Hence the end of the compression line, the lead or admission line, and the commencement of the steam line, will appear near the centre of the crossed diagram, while the extensive movement of the indicator drum at this point results in a lateral spreading of the lines, so to speak, which renders their actual formation much more evident than with the ordinary card. In Fig. 72 the full line is the direct diagram obtained from the low-pressure cylinder of a marine engine, while the dotted lines show the crossed diagram produced by attaching the cord to the lever of the high-pressure cylinder, the crank of which was 90° ahead of the low-pressure. The compression line $a\ b$, when developed in the crossed diagram, has now the more distinctive outline shown by $a'\ b'$, while the vertical admission line $b\ c$ becomes $b'\ c'$.

A somewhat curious crossed diagram is obtained when the cut-off occurs later than at half-stroke, both the steam and exhaust lines doubling back upon themselves, as shown

in Figs. 73 and 74, in which similar lines are similarly lettered.

Diagrams Indicating Defects in Engine Design, Leakage, Etc.

In view of the detailed consideration which has been given in Chapter II. to the defects caused by faulty engine design and construction, it will here be unnecessary to give more than a few general examples of actual diagrams bearing upon the points at issue. The closeness with which the admission pressure approximates to the boiler pressure is an important factor in the economical use of steam. In many cases, indeed, the alterations and adjustments which it is possible

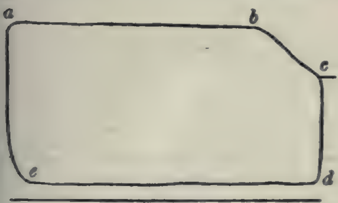

Fig. 73.

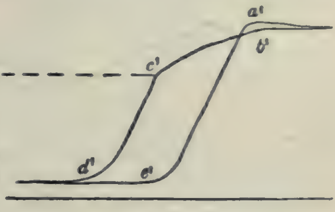

Fig. 74.

to effect in the subsequent events of the stroke are insignificant in comparison with the improvement which can be realised by securing a higher initial pressure and a well-sustained steam line. Hence more attention should be given to the provision of well-clothed steampipes of sufficient size, as free from bends as possible, and thoroughly drained of entrained water and that due to condensation. The engine stop and throttle valves are frequently so designed as to offer an unnecessarily large resistance to the flow of steam, while slide valves are often so arranged that the free ingress of the steam to the cylinder is needlessly impeded, this being more particularly evident in some forms of piston valves. But even when the valves are set correctly and the ports are of ample area, cases are sometimes met with in which the highest pressure attained during admission unaccountably falls short of the boiler pressure, and to a very

marked extent. Thus, in the diagram shown in Fig. 75, taken from the high-pressure cylinders of a new marine engine, it is seen that while one card indicates a satisfactory steam distribution, that from the opposite end of the cylinder exhibits a considerable amount of wiredrawing during admission, causing the steam line to fall much below that attained in the other diagram. An explanation of this unequal steam distribution was not forthcoming until the engine was overhauled, when on lifting the cylinder cover it was found that the part of the latter which was let into the cylinder barrel partially covered the steam orifice, thus impeding the steam flow to that end of the cylinder. Upon chipping away the obstruction, similar cards from both ends were obtained.

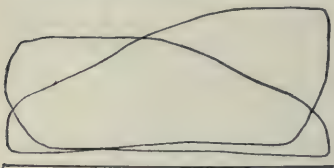

FIG. 75.

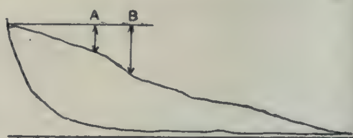

FIG. 76.

The difficulty of maintaining the admission pressure becomes increasingly marked as the speed of revolution of the engine is higher, and hence diagrams from high-speed automatic cut-off engines generally show a sloping steam line. The diagram shown in Fig. 76, which will serve as an example, was taken from a small engine of the type named when running at 260 revolutions per minute. The pressure at the point A, where cut-off commences, shows a fall of about 16lb. from that initially attained; subsequently the pressure falls still more rapidly, until the valve closes at B, this being due to the rapidly diminishing port opening in conjunction with the increasing piston velocity. It will be noted that the release occurs too early in the stroke, but even with the very low terminal pressure resulting it is still necessary to employ a full degree of compression in order to fill the clearance space with steam at a sufficiently high pressure to secure smooth running. From this it will be seen that any attempt to improve the steam line by adopting

larger port openings will have the effect of increasing the clearance volume, and hence will necessitate the still earlier commencement of the compression period, thus in all probability more than neutralising the advantage gained during admission. Similar examples of wiredrawing during admission are shown in the diagrams from locomotives at short cut-offs and high speed (see Fig. 57, page 55).

Leakage past the piston during admission probably occurs to a far larger extent than is usually suspected; but in most cases it is practically impossible to detect it by the indicator, since the escaping steam finds a free outlet through the exhaust opening on the other side of the piston, and hence has little or no observable effect upon the back pressure line of that diagram. When sufficient compression is used in a single-valve engine to raise the pressure near the end of the stroke well above that on the opposite side of the piston, leakage is sometimes indicated by a more or less marked drop in the compression curve (see Fig. 43). In such cases it may be reasonably inferred that leakage will similarly occur during the ensuing admission period, and result in a much greater loss, since the forward pressure will be much higher and the back pressure lower than during the previous compression period. A rounding of the admission curves of the diagram, in conjunction with a distorted compression curve, may in general be taken as confirmatory evidence of piston leakage, always provided, of course, that the rounding referred to is not due to late admission. However, the only trustworthy method of detecting leakage is by removing the cylinder cover, blocking the piston at various points in the stroke, and testing with full pressure steam.

The effect of piston and valve leakage upon the expansion curve has already been dealt with in a previous chapter (page 33), but it is desirable here to point out the effects separately produced in the diagram by the various forms of leakage. These may be described as (1) leakage past the valve, which will tend to *raise* the expansion, release, exhaust, and compression lines; (2) leakage past the piston, tending to cause the expansion and release lines to *fall* unduly, and the exhaust and compression lines to *rise* above their normal positions; and (3) leakage of air past the piston-rod gland

in condensing engines, which will affect one card only, and tend to raise the exhaust and compression lines owing to the impaired vacuum resulting.

These are the effects which would be produced by the separate influences named; but at best they can only be regarded as suggestive, inasmuch as varied combinations of the several forms of leakage may easily occur, and indeed are not infrequently met with.

In engines of the four-valve type, leakage may sometimes assume very serious proportions. In Corliss engines the valve seat may be so worn as to leave a ridge at the end of the valve travel, and hence a readjustment of the gear may

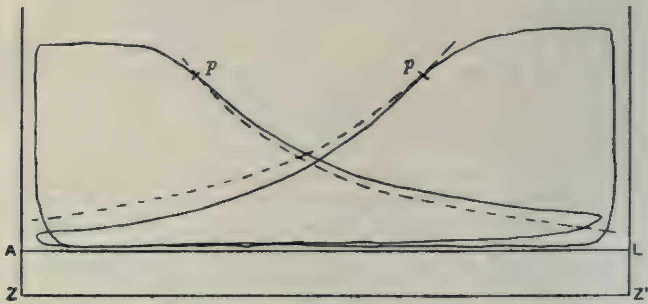

Fig. 77.

cause the valve to lift, and leakage to occur, as it moves on to the hitherto unworn part of its seat.

An example of leakage of both steam and exhaust valves is given by the cards, shown in Fig. 77, from a Corliss engine 22in. in diameter with a 5ft. stroke and running at 60 revolutions per minute. The point of cut-off is not as sharply defined as is usual in engines of this type—this in itself suggesting leakage; but by drawing hyperbolic reference curves through the points p and p', where the valve is completely closed, a considerable departure from the standard expansion curves at once becomes evident. In the left-hand diagram the actual expansion curve rises materially above the theoretical. This may be due wholly or in part to leakage of the steam valve during expansion,

or to re-evaporation of steam condensed during admission. The latter cause is, however, insufficient to account for the marked excess of pressure at the point of release, since at a piston speed of 600ft. per minute the re-evaporation ordinarily obtained in engines of this class, size, and cut-off, is not more than 3 or 4lb. above the theoretical. Hence it may be safely assumed that a marked leakage of the steam valve is taking place at this end of the cylinder. In the right-hand card the conditions are reversed, the weight of steam in the cylinder at release being much less than that due to the standard expansion curve. It is obvious that in this case no question of re-evaporation can enter, but it is equally evident that the real discrepancy is greater than that shown in the diagram by the amount which is to be allowed for the influence of the re-evaporated steam. The conditions here shown undoubtedly suggest leakage of the exhaust valve as the most probable cause of the loss of pressure shown. A similar effect would be produced by leakage past the piston; but this would affect *both* expansion curves, and hence if it took place in the case under notice it would indicate that the leakage in the left-hand diagram is much greater than the amount shown.

Among other faults shown in these cards is the late opening of the exhaust valve in the left-hand diagram, with a consequent material increase in the back pressure as compared with the card from the opposite end.

The most important effect of late opening to exhaust is its influence in raising the back pressure line during the early part of the return stroke. It is true that with ample port area and large exhaust pipes the back pressure soon attains its lowest limit, but in most cases it will be found that a slight amount of additional work done near the end of the forward stroke (work which is largely absorbed in engine friction) is much more than compensated for by the loss during the return stroke.

A further consideration which is not without weight in high-speed engines is the higher compression pressure which a late release entails, as obviously the pressure available for cushioning purposes is the difference between that due to the compression on the one side of the piston and that due to the forward pressure on the other.

The utilisation of exhaust steam for heating purposes inevitably detracts from the efficiency of the engine by the increase in the back pressure due to the resistance of the heating system. With large and well-arranged pipes, however, the back pressure in ordinary cases need not exceed 4 or 5lb. per square inch. When the loss occasioned much exceeds this amount, the economy of the method as against using boiler steam may be open to question, and should certainly be investigated.

Occasionally diagrams from condensing engines show a constantly rising back pressure line, as in Fig. 78. This may be due to the supply of injection water being insufficient to prevent the accumulation of pressure in the condenser. More often, however, it suggests a defective valve in the air-pump bucket. When the latter defect is such as to lead to an intermittent action, the back pressure line may assume a more or less undulatory character.

In horizontal double-acting air-pumps, leakage of the air-pump gland is often the cause of a deficient vacuum, as is also leakage past the bucket owing to defective packing. Air may find its way direct into the condenser by entering through the glands of the piston rod or valve rod, or through drain cocks and other cylinder fittings. In marine engines further causes of reduced vacuum are found in leakage past the low-pressure cylinder relief valves and badly-packed expansion joints, and the leakage from the glands and drains from the auxiliary engines and the connected piping communicating with the main condenser.

Diagrams from Underloaded and Overloaded Engines, Friction Diagrams, Etc.

It is a well-recognised fact that for every engine running at a fixed speed there is a definite load under which economy in the use of steam attains the maximum. A moderate departure from the "economical load" in either direction does not, as a rule, lead to more than a moderate increase in the steam consumption, but a marked deviation from the best conditions of working invariably involves a serious loss of economy, this being more particularly the case when the engine is underloaded.

Underloaded Engines.—In a general way it may be said that steam is used most economically when the mean effective pressure is greatest for a given (absolute) terminal pressure, the latter being determined by dividing the absolute initial pressure by the ratio of expansion employed. From this it follows that theoretically the cut-off should be such as will give a terminal pressure equal to the back pressure. In practice, however, it is found perferable to arrange for a nominal terminal pressure of about 2 to 6lb. per square inch above the atmosphere, or about 17 to 22lb. absolute in the case of non-condensing engines, and from 6 to 12lb. absolute in the case of condensing engines. These figures can only be considered as roughly indicating the advisable limits between which the nominal terminal pressure should

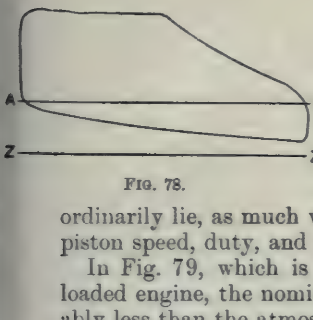

FIG. 78.

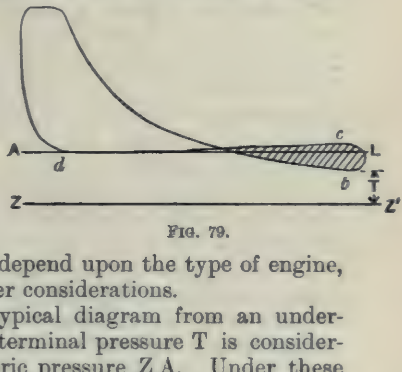

FIG. 79.

ordinarily lie, as much will depend upon the type of engine, piston speed, duty, and other considerations.

In Fig. 79, which is a typical diagram from an underloaded engine, the nominal terminal pressure T is considerably less than the atmospheric pressure Z A. Under these circumstances the opening of the exhaust valve at *b* is followed by an inrush of air into the cylinder, the pressure ultimately rising to *c* in the return stroke, from whence the diagram is completed, as shown. As the area included in the shaded loop represents work done by the engine upon the atmosphere, the amount of work so absorbed is to be deducted from that represented by the area of the upper part of the diagram, the resulting difference showing the low mean effective pressure required to drive the load under the conditions here prevailing. But beyond the loss of efficiency due to the useless work done, there is to be considered the loss due to cylinder condensation, which, when the cut-off

occurs so early in the stroke, becomes a very serious item. Irregular running due to the wide variation of pressure on the piston is another, although a comparatively minor, disadvantage attending light loads. Assuming that structural alterations of the cylinder are inadmissible, two methods of dealing with lightly-loaded engines suggest themselves. These are (1) reducing the initial pressure by lowering the boiler pressure, or, preferably, either by throttling or by using a pressure-reducing valve; and (2) reducing the speed of the engine, and hence increasing the mean effective pressure. Much will depend upon the nature of the load. If the latter is fairly constant, reduction of speed is to be advised if at all practicable, or failing that, a combination of both methods. If, however, the load fluctuates considerably, as with engines in electric generating stations, the reduction of the initial pressure is the only plan admissible. This fact has been recognised by some of the builders of Corliss engines for electric generating plant, who arrange the governor to operate a throttle valve, at light loads only, in order to obviate the losses otherwise incurred with underloaded engines. In locomotives, early cut-off almost invariably occurs at high speed, but in this case the loss above referred to is very conveniently avoided by the heavy compression which is a feature of steam distribution by the link motion. Excessive compression in ordinary stationary-engine practice would, however, lead to heated journals and unsteady running, and hence is not to be included among the expedients for combating the evil effects of light loads.

It should be noted that in non-condensing engines a reduction in boiler pressure is not calculated to promote economy, unless the terminal pressure is lower than that due to the atmosphere. Further, alterations in engine speed should be considered in regard to the ability of the fly-wheel to effect the necessary closeness of speed regulation at the reduced velocity contemplated.

Overloaded Engines.—The loss of economy is distinctly less with overloaded than with underloaded engines, as in the former case the influence of cylinder condensation does not appear as a prejudicial factor, the loss being almost entirely due to the inability to realise the full benefit of expansive working owing to the lateness of the cut-off.

A case in point is afforded by Fig. 80, which is a diagram from a Corliss engine with a cylinder 18in. in diameter and 42in. stroke, running at 75 revolutions per minute. As will be seen, the cut-off occurs late in the stroke, and hence the pressure at the point of release is much higher than desirable. The suggestive remedies in this case are (1) increasing the speed, (2) using steam of a higher pressure, and (3) adding a condenser. Fig. 81 shows the diagram obtained after the speed had been increased to 90 revolutions per minute, and, as will be seen, represents a very much closer approximation to the economical load of the engine, this being further shown by the reduction in the water consumption from 25·6 to 21·5lb. per I.H.P. per hour. A limitation to the increase of speed in Corliss engines lies in

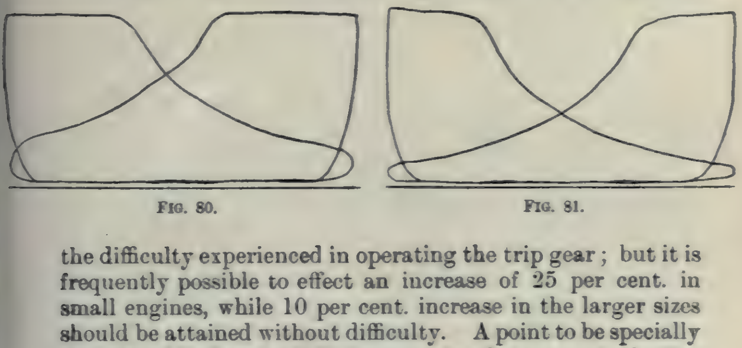

FIG. 80. FIG. 81.

the difficulty experienced in operating the trip gear; but it is frequently possible to effect an increase of 25 per cent. in small engines, while 10 per cent. increase in the larger sizes should be attained without difficulty. A point to be specially considered is the probable increase in the amount of wire-drawing at the ports, for unless these latter are of ample area, the loss of pressure at the higher speed may go far to off-set the advantage otherwise gained. Due regard must also be paid to the effects of increased wear, and the ability of the parts to withstand this; the stresses due to the higher speed must also be taken into consideration.

The possibility of using higher steam pressure depends, of course, mainly upon the extent to which the boiler pressure may be increased. Usually, overloaded engines are the outcome of additions which have been made to a manufacturing plant from time to time, and hence, unless the boiler had originally a large margin of strength, it

would be improbable, owing to deterioration with years of working, that any increase of pressure would be permissible. When, however, new boilers are installed, or a higher steam pressure is otherwise available, this method of relieving an overloaded engine will in general be attended with most satisfactory results, always provided the several parts of the engine, and particularly such wearing surfaces as the crank and crosshead pins, shaft necks, etc., are calculated to withstand the increased load.

Adding a Condenser.—By the addition of a condenser to a non-condensing engine it is generally possible to reduce the back pressure from some 16lb. absolute to about 4lb. absolute, thus increasing the mean effective pressure by about 12lb. per square inch. In the case of an overloaded engine this additional power enables the steam to be cut off much earlier in the stroke, hence giving a closer approach to the economical load of the engine. Against the gain in pressure thus obtained there is to be set the power required to work the air-pump; while in cases where the exhaust steam was previously used to heat the feed water, or for other purposes, these losses should also be taken into account. The cost of condensing water, if the supply has to be purchased, and, of course, the interest on first cost, depreciation and upkeep of the addition to the plant, are all points to be considered in this connection.

When the loss of the heated feed water is the main objection to adding a condenser, the advisability of using an economiser should be considered, as by this method of utilising the waste gases from the boiler, a much hotter feed can generally be obtained than is possible with exhaust steam heaters. For small and medium powers, a convenient means of taking advantage of the gain due to condensation is offered by the improved forms of ejector condensers now available; while for larger plants some form of independent condenser will usually be found preferable.

A typical example of the gain due to adding a condenser is shown in the superposed diagrams, Fig. 82, taken from an engine with a cylinder 20in. in diameter, 40in. stroke, and running at 70 revolutions per minute. The mean effective pressure is 43lb. per square inch in each diagram,

but by running condensing, the cut-off is reduced from $\frac{1}{3}$ to $\frac{1}{5}$ of the stroke, with a marked economy of steam. The loss by cylinder condensation will be increased both by reason of the earlier cut-off and the greater range of

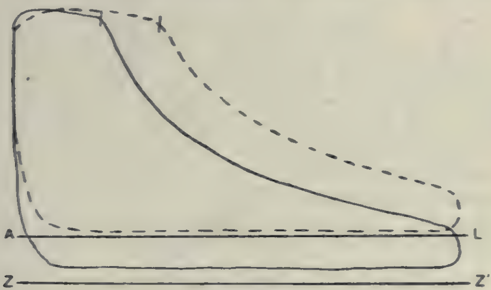

FIG. 82.

temperature in the cylinder; but making due allowance for this and other prejudicial factors already referred to, a substantial saving would undoubtedly be effected.

Friction Diagrams.—Diagrams taken from unloaded engines, and therefore showing the power required to overcome the friction of the engine under these conditions, may evidently be regarded as exaggerated cases of underloading.

FIG. 83.

FIG. 84.

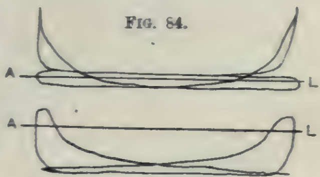

FIG. 85.

Frequently loops of negative work appear at one or both extremities of the diagram, especially when the necessarily early cut-off is effected by link motion gear. Fig. 83, which gives a case in point, shows diagrams taken from the high-pressure cylinder of a marine engine of 450 I.H.P. The left-hand card gives a double loop, the upper one due to

excessive compression, and the lower to excessive expansion. In many cases friction diagrams from compound engines show an excess of negative work in one or other of the cylinders. Figs. 84 and 85 afford an illustration, the negative work loops in the high-pressure diagrams being much in excess of the positive work areas, leaving an unduly large amount of work to be done in the low-pressure cylinder.

In engines fitted with releasing valve gear, friction diagrams similar to Fig. 86 are often obtained, the cut-off virtually taking place at the commencement of the stroke, and practically leaving the work to be done by the expansion of the steam in the clearance space. Frequently, in friction diagrams from high-speed automatic cut-off engines, undu-

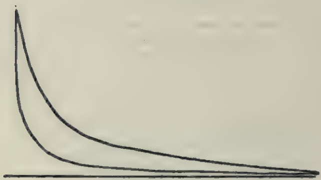

FIG. 86.

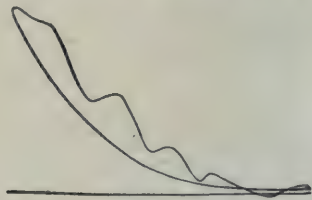

FIG. 87.

lations in the expansion line (Fig. 87), due to the momentum of the indicator mechanism, render it difficult to calculate the small area of the card with the necessary degree of accuracy, and hence but little reliance can be placed upon values obtained from such diagrams. Undoubtedly the most satisfactory method of ascertaining the work absorbed by engine friction is to determine both the brake and indicated horse-power, the difference in the two amounts giving a more accurate measure of the work absorbed than can be obtained by means of friction diagrams. In large engines only the latter method is available; but the fact should be kept in view that the percentage of actual work expended upon engine friction when the engine is fully loaded is somewhat larger than a comparison of the friction and full-load diagrams would suggest.

CHAPTER IV.

DIAGRAMS FROM COMPOUND ENGINES.

INDICATOR diagrams from compound, triple-expansion or quadruple-expansion engines do not differ essentially from those obtained from simple engines, and hence what has been said regarding the latter is, with obvious modifications, equally applicable to cards from multi-cylinder engines. More or less marked departures from the ideal simple diagram are, however, almost invariably found in the *separate* cards from compound engines, but it is to be noted in this connection that the comparison should rather be instituted with the *combined* cards, or otherwise, with the upper line of the high-pressure and the lower line of the low-pressure diagrams.

In the majority of instances the piston strokes are of the same length, and hence for such cylinders as are arranged in tandem order it is usually possible to drive both indicators from the one reducing gear. Under these circumstances the diagrams from the two cylinders are most conveniently taken of the same length, but with the pendulum types of reducing gear it is obviously possible to arrange the lengths of the diagram from the high- and low-pressure cylinders to be proportionate to the volumes of the respective cylinders. This is of some advantage in the subsequent combining of the cards, but unless the proportionate reduction is accurately effected (and this is not readily accomplished), the results are entirely misleading, and worse than useless for the purpose named. The method is therefore not generally to be recommended.

In taking cards from compound locomotives and high-speed single-acting engines, it is customary to employ springs of the

same scale. Hence when a pair of such diagrams are taken from an engine having no receiver, the action of the steam in its passage through the two cylinders can be well shown by placing the two cards in their correct positions relative to the atmospheric line.

The diagram Fig. 88, from a high-speed compound engine, is a good example, the drop from the back pressure line of the upper card to the steam line of the lower being comparatively slight. In engines provided with a receiver, however, such as is more generally necessary in multi-cylinder engines, no such representation of the continuous action of the steam can be obtained without combining the several cards.

With the exceptions above noted, the low-pressure diagrams of compound engines are usually taken with a

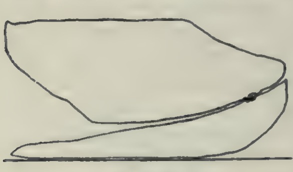

FIG. 88.

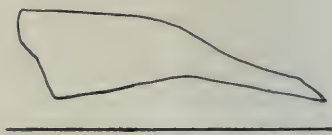

FIG. 89.

low-scale spring in order to show the action of the steam more conveniently. Similarly, for the cards from triple and quadruple engines, a spring of different scale is used for each cylinder, and although this method complicates the operation of combining the diagrams, it is otherwise convenient, inasmuch as it gives a series of cards of suitable proportions and of such a size as will allow the area, and hence the mean effective pressure, to be determined with accuracy. Whenever possible, it is preferable to use a spring for the higher pressure cards which is a convenient multiple of that used for the low pressure. By this means the aggregate pressure at any point in the stroke is much more readily obtained, while much time and labour may be saved when combining the cards, as will presently appear.

In engines without a receiver—tandem engines or side-by-side cylinders with cranks at 180°—the steam exhausts directly from the high-pressure cylinder into the next

following and larger cylinder. Hence the consequent expansion causes the back pressure line of the small cylinder to fall, as indicated in Fig. 88, until compression commences. A similar fall necessarily takes place in the steam line of the low-pressure card, the space between these two lines representing frictional and other losses incurred in the transference of the steam. On the other hand, cards from engines provided with a receiver do not exhibit this feature, each of the diagrams conforming more closely to a simple-engine card, especially if the receiver is of suitable volume. Usually the back pressure line of the high-pressure card will rise gradually towards mid-stroke, when the cranks are at right angles or otherwise, until the admission to the low-pressure cylinder commences, after which the pressure will fall. Fig. 89 shows a particularly bad case of insufficient receiver volume and small port areas. In the low-pressure card the steam line will follow the pressure variation in the receiver, but if the receiver is not unduly small the wiredrawing effect, although always existent, should not be specially marked. Similar variations will be seen in cards from triple and quadruple engines, the precise form depending upon the sequence and relative angles of the cranks, capacities of receivers, cylinder ratios, etc. Thus if the cut-off in the intermediate (or low-pressure) cylinder is shortened by separate linking up, the initial pressure in that cylinder will be increased, and as the receiver space (which more usually comprises the volume of the connecting steampipe, etc.) is limited, a more sloping steam generally results, the effect being less noticeable as the receiver capacity and the port areas are larger. If the cranks are arranged in such sequence that the discharge from the first series coincides approximately with the opening to steam of the next cylinder in the series, a fairly well-sustained steam line may often be obtained in the card from the latter cylinder, although in any case it will be evident that in all cylinders after the H.P. the conditions are really equivalent to the expansion of a large volume of clearance steam.

Unequal Distribution of Power in compound and multi-cylinder engines, which it is highly desirable to avoid in order to secure smooth running, can generally be rectified by separately modifying the cut-off in one or more of the

cylinders whenever the means for this individual adjustment is provided. In a two-stage compound engine, cutting off earlier in the H.P. cylinder will diminish the total power developed, slightly decrease the mean effective pressure (M.E.P.) in the H.P. cylinder, while materially reducing the M.E.P. in the L.P. cylinder. Hence, relatively, a much larger proportion of work will be done in the H.P. cylinder under these conditions.* On the other hand, cutting off earlier in the L.P. cylinder alone will reduce the M.E.P. in the H.P. cylinder, while increasing the M.E.P. in the L.P. cylinder, both effects tending to throw more work on to the latter cylinder. The total power developed will, however, be unchanged, as will also the terminal pressure, since the latter depends upon the cut-off in the H.P. cylinder and the volume of the L.P. cylinder. In triple-expansion engines, cutting off earlier in the H.P. cylinder reduces the whole power developed, increases relatively the M.E.P. in the H.P. cylinder, and lowers the M.E.P. in both the intermediate pressure (I.M.P.) and L.P. cylinders.

Cutting off earlier in the I.M.P. cylinder lowers the M.E.P. in the H.P. cylinder, and increases the M.E.P. in the I.M.P. cylinder, but does not materially influence the M.E.P. in the L.P. cylinder.

Cutting off earlier in the L.P. cylinder usually has little or no effect upon the M.E.P. in the H.P. cylinder, reduces the M.E.P. in the I.M.P. cylinder, and increases the M.E.P. in the L.P. cylinder. Cutting off earlier in both the I.M.P. and L.P. cylinders will increase the power developed in the L.P. cylinder and reduce the power of the H.P. without materially affecting the M.E.P. in the I.M.P. cylinder.

Combining Diagrams.—Although the successive use of steam in two or more cylinders is generally advantageous from an economic point of view, the transference of the steam through its several stages involves losses which it is obviously desirable to reduce as far as possible. The nature and extent of these losses are approximately shown by combining the diagrams from the several cylinders in such a manner that the resultant figure shows the con-

* From this it follows that in underloaded compounds the greater part of the work is generally performed in the H.P. cylinder, while when the load is excessive the conditions are reversed.

tinued course of the steam through the engine, but so rearranged as to represent in effect an equivalent simple diagram of the pressure which may be supposed to have acted continuously upon the L.P. piston only.

Opinion is by no means unanimous as to the correct method of locating the diagrams in a combined card. Much will depend upon the ultimate object of the combination. The method most usually pursued, probably by reason of its simplicity, consists in (1) reducing the diagrams to a uniform scale of pressure, (2) reducing the diagrams to a uniform scale of cylinder volumes, and (3) placing the modified cards in their correct positions relative to the atmospheric line, each card being displaced from the zero line of volume by an amount equal to the percentage of clearance in its cylinder, but taken on the common scale of volumes adopted. In tandem engines the crank-end card of the H.P. should be combined with the head-end card from the L.P., unless a mean card from the two ends of each cylinder is first prepared, a plan frequently followed in combining cards from receiver engines.

As an illustration of the method, the diagrams Figs. 90 and 91 from a compound tandem engine are shown combined in Fig. 92, the plan of procedure being as follows:—Divide both of the original cards into a convenient number of equal parts, as 10, ruling ordinates as shown. Set off from the first or end ordinate the clearance volume, and draw the clearance line or ordinate of zero volume. In the example taken, the H.P. clearance is 1·03cub. ft., and the L.P. clearance 2·28cub. ft., corresponding to 7·25 per cent. and 4·5 per cent. of their respective piston displacements. The cylinder diameters are 23 and 43in., giving piston displacements of 14·43cub. ft. and 49·8cub. ft. respectively.

To combine these cards, draw a line 10 or 12in. long (O V, Fig. 92), representing the line of zero pressure, and also a convenient scale of volumes. Draw the ordinate of zero volume O P, which will also serve as a scale of absolute pressures. From O set off O E equal to the L.P. clearance volume (2·28cub. ft.) on the scale of volumes. From E lay off a length E B equal to the L P. piston displacement, 49·8cub. ft. as shown. Whenever possible the scale selected for the pressures should be the same as that of the original

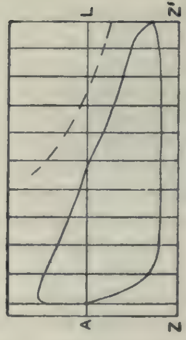

FIG. 91.

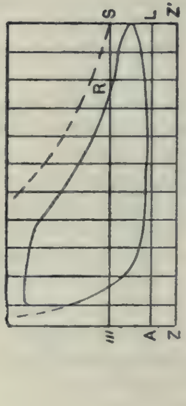

FIG. 90.

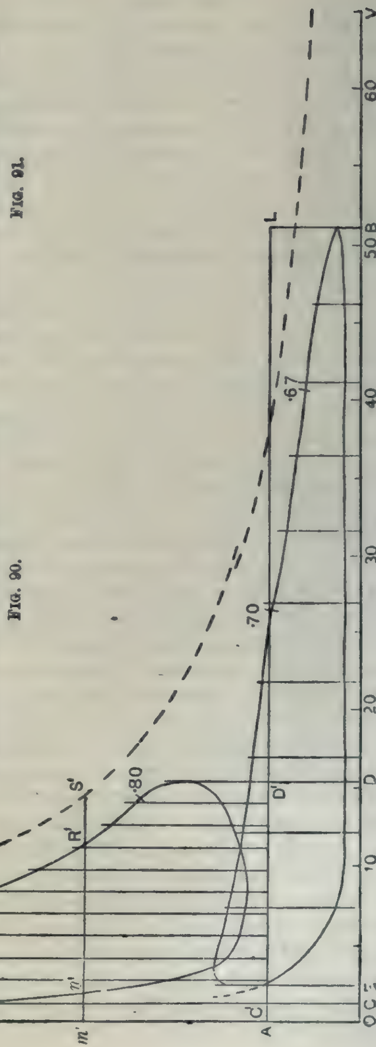

FIG. 92.

L.P. card, since pressures on the latter can then be directly transferred to the new diagram; otherwise, some convenient multiple of the original scale, say 1·5, should be chosen. The scale of Fig. 90 is 40, and that of Fig. 91 is 10.* The latter is therefore used for the combined card, and a scale of pressures drawn in accordingly. Divide E B into the same number of equal parts as the L.P. diagram—in this case 10; draw in the ordinates and mark off on these the absolute forward and back pressures from the corresponding ordinates of the original L.P. card. In this way a series of points will be obtained through which the elongated L.P. card can be drawn.

For the H.P. card, set off from O P the H.P. clearance volume O C = 1·03cub. ft., and draw the ordinate C C′. From the latter set off the H.P. piston displacement (14·24ft.) as C D, or more conveniently as C′ D′, on the atmospheric line. Divide C′ D′ into 10 equal parts and draw the corresponding ordinates. As the scale of pressures of the original H.P. card is 40, the heights representing the absolute pressures therein will require increasing fourfold before they are transferred to their corresponding ordinates in Fig. 92, after which the modified H.P. card can be drawn in as shown. When the scales of the original and combined cards are not in a simple ratio, a pair of proportional compasses furnish a convenient means for converting and transferring the ordinate values, but otherwise it is preferable to mark the pressures on the original card by the scale of the spring used, afterwards laying them off on the combined card. The process of combining the cards is completed by drawing in a reference curve of expansion, a saturation curve or a rectangular hyperbola being generally employed. Most frequently this reference curve is drawn in one unbroken length, but this plan is only warranted when the same quantity of steam is expanding in all the cylinders, or, in other words, when the same weight of compressed or cushion steam is retained in each cylinder. This is a coincidence not generally met with, as usually less cushion steam is retained in the L.P. cylinder. Nevertheless, provided the limitations of the method are fully recognised, a

* The diagrams in the illustration are on a reduced scale.

curve drawn either through the actual or the assumed point of cut-off supplies an amount of useful information. When the weight of steam passing through the engine can be ascertained, the more correct method of comparison is to draw in on each of the original cards the saturation curve corresponding to the known weight of feed. These curves are then transferred to the combined diagram by marking off on any horizontal, as m' S′ (Fig. 92), a length R′ S′ bearing the same ratio to m' R′ as does R S to m R (Fig. 90).

The two saturation curves having been transferred to the combined diagram, it becomes an easy matter to determine the relative amounts of steam and water present in each cylinder—in other words, to determine the *dryness fraction.* Thus, if in Fig. 92 a horizontal m' S′ is drawn passing through both the expansion and compression lines of the diagram, then m' n' will represent the volume of the cushion steam, n' R′ the volume of working steam present, and R′ S′ the amount of feed existing as water. Hence, since of the whole, m' S′, the proportion existing as steam is m' R′, the ratio $\frac{m' R'}{m' S'}$ represents the *quality* of the steam or dryness fraction at the point R′ selected in the H.P. stroke. In cases where the actual compression curve does not extend sufficiently, a rectangular hyperbola is to be drawn continuing the curve as shown by the broken lines in Fig. 90. If the compression line is small or indefinite, a hyperbola should be drawn through a point just after the exhaust valve is completely closed. Pursuing this method of investigation, it will be found that the dryness fraction at cut-off in the H.P. diagram, is 0·72, and at release 0·80, showing that a portion of the steam liquefied during admission had re-evaporated during expansion. On the other hand, in the L.P. cylinder the dryness fraction at cut-off is 0·70, and at release it has fallen to 0·67, showing a further slight condensation.

The method of combining cards from triple and quadruple engines being precisely similar to the foregoing, further illustrations are unnecessary. It may be added that when, as is frequently the case, any one or more stages of the expansion are divided among two equal cylinders, the

combined clearances and working volumes are to be taken for each stage so divided, the corresponding pressure variation throughout the stroke being the mean of those given by the two cards.

Another method of combining diagrams which deserves attention is one in which the variable amount of cushion steam is eliminated, thus allowing a comparison with a standard area bounded by a continuous saturation or other expansion curve. To transform the diagrams in Fig. 92, the same scale of pressures is used (O P, Fig. 93), but the corresponding volumes are set back by the amount assumed to be occupied by the cushion steam. Thus, for the pressure O f' (Fig. 93) corresponding to O e in Fig. 92, a length $f' g' = f g$ (f being a point on the produced compression curve) is set off from O P giving the point g' in the new card. Similarly, $g' h'$,

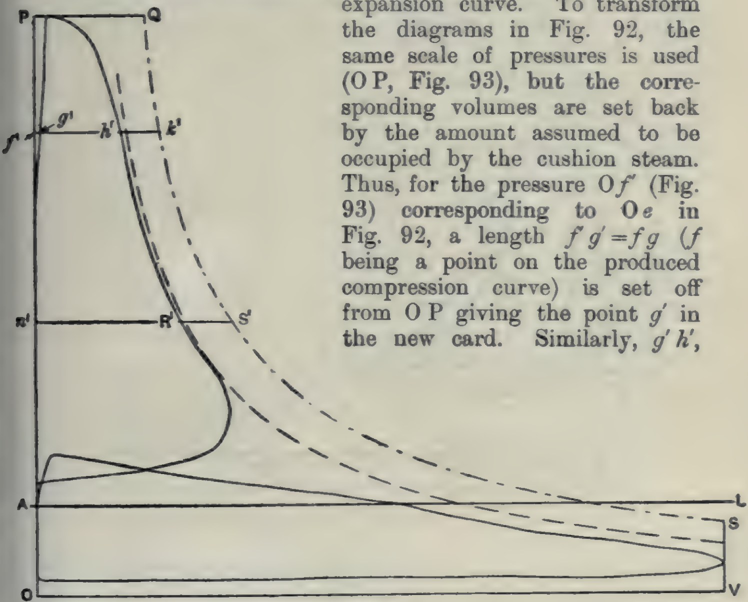

FIG. 93.

equal to $g h$, gives the point h' in the transformed expansion curve. Proceeding in this way, the new diagrams are drawn as shown in Fig. 93, and if the amount of cylinder feed per stroke is known, this may be set off at P Q and a saturation or other curve drawn. Then at any point as n' the length n' R$'$ represents the volume of working steam, and R$'$ S$'$ the volume condensed in the form of water. The triangular areas between the two cards and O P repre-

sent work lost in filling the clearance spaces with boiler steam, the spaces between the cards, the losses incurred in transferring the steam—such as wiredrawing, receiver drop, radiation, etc.,—and finally the ratio of the area of the two cards (the effective work area) to the whole area O P Q S V is a measure of the efficiency of the steam—*i.e.* the proportion of work done to that theoretically possible from the same weight of steam expanding under whatever condition is assumed in constructing the curve Q S.

When the cylinder feed is not known, it is desirable to con-

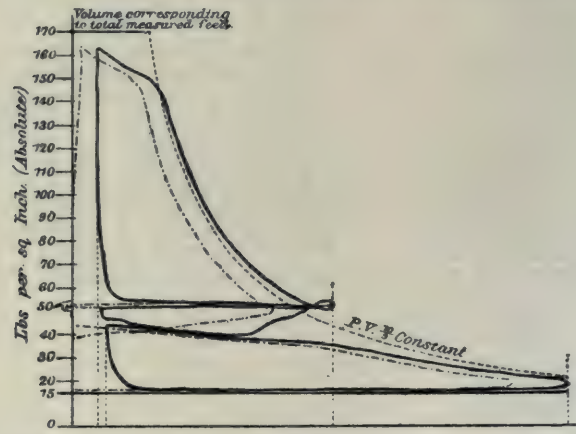

FIG. 94.

struct a saturation curve just touching the most prominent point in the combined diagram—in other words, through the point of least wetness. This will then represent a curve of *uniform wetness*, and will afford a means of tracing the relative condensation of the steam as it passes through the engine.

In single-acting stage expansion engines following the Cornish cycle, and in which the steam after acting upon the upper side of the piston is transferred to a receiver beneath the piston, the card from this receiver is to be incorporated in forming a combined diagram. A combined

diagram from such a compound engine * is given in Fig. 94, the centre card being that obtained from the intermediate receiver. The same cards, when the cushion steam is eliminated, are shown in broken lines, the expansion curve being an adiabatic corresponding to the total measured feed admitted to the engine.

In combining diagrams it will often occur that portions of the cards overlap one another—as, for example, is the case with those shown in Fig. 92. Upon consideration it will be seen that this is due to the foreshortening of the H.P. card, from which it follows that points on the same ordinate, but on the steam line of the L.P. card and the back pressure line of the H.P. card respectively, no longer represent the coincident positions of the two pistons. Hence when it is desired to examine more particularly the losses due to wire-drawing between the cylinders, a convenient plan is to construct a diagram in which such portions of each card as are formed while connection with the receiver is open are placed in their correct relative position. For this purpose it is customary to use a vertical scale of pressures, and a base scale representing the degrees passed through by any one crank. Upon transferring such portions of the several cards as represent the periods during which intercommunication exists between the cylinders, the extent of the losses incurred may be very readily determined.

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* Proceedings Inst. Civil Engineers, vol. xciii.

CHAPTER V.

DIAGRAMS FROM GAS AND OIL ENGINES.

Gas Engine Diagrams.

WHAT may be termed the *theoretical indicator diagram* of a gas engine working on the four-stroke, "Beau de Rochas," or "Otto" cycle now so largely adopted, is shown in Fig. 95. Here A L represents the line of atmospheric pressure, A P the ordinate of zero volume, B C the stroke, and A B the proportionate clearance volume, the amount of the latter varying considerably, but approximately from 50 per cent. of the piston displacement in the early types, to 20 per cent. or less in modern engines. Taking the four successive strokes in order, B C marks the admission of the explosive mixture at atmospheric pressure. During the return stroke the charge is compressed, the consequent rise in pressure being indicated by the adiabatic curve C D. At D the charge is ignited, D E marking the instantaneous rise of pressure due to the explosion. During the working stroke, the gases expand and the adiabatic curve E F results. At F, opening to exhaust occurs, the pressure falling instantly to C, and the products of combustion are expelled during the return stroke C B. Hence, of the four strokes, the pressure during the first and fourth does not vary from that due to the atmosphere. In the second stroke work is done on the charge by the stored energy of the motor, this being represented by the area C D B; while during the third stroke work is done by the expanding gases to an amount shown by the area B E F C. Hence, deducting the negative from the positive work, there remains the area D E F C as a measure of the net or effective work done in the complete cycle.

The Gas Engine Diagram in Practice.—It is tolerably evident that the conditions required for perfect working are incapable of realisation in practice, more particularly in regard to the expansion taking place without involving an interchange of heat between the charge and the water-cooled cylinder walls as assumed in the foregoing. Moreover, in practice the explosion does not take place instantaneously, nor is combustion completed until some portion of the stroke has been accomplished. From these and other practical modifications it results that the actual gas engine diagram more nearly resembles the form shown in Fig. 96, except that the lower lines are not usually shown so definitely.

Commencing at B as before, it is seen that owing to wire-drawing during admission, the line B C falls somewhat

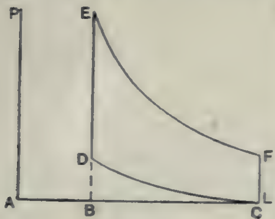

FIG. 95.

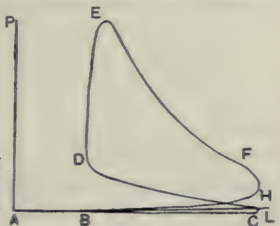

FIG. 96.

below the atmospheric line, showing that a slight vacuum is formed in the cylinder during this stroke. During the compression C D the conditions are such that often close approximations to the adiabatic curve may be obtained. If the engine is worked with a fairly hot cylinder, the curve will rise above the theoretical; if the cylinder is effectually cooled, it will fall below. That the explosion of the charge is not effected instantaneously is shown by the manner in which the line D E leans inward, indicating that the piston stroke had well begun before the highest pressure was attained. This, however, will be influenced by the amount of "firing lead" adopted. Some time is occupied in completing the process of combustion resulting in the formation of the rounded corner shown at E. A more or less marked deviation from the adiabatic curve is usually met with in the expansion curves of actual diagrams, the absorption of

heat by the cylinder walls, and the rate at which combustion of the charge proceeds being among the principal of the somewhat complicated influences which decide the ultimate form of this curve. The exhaust valve is opened before the end of the stroke is reached, with the object of effecting as rapid a fall of pressure as possible before the return stroke commences, and when the point H is reached the pressure is not greatly in excess of that due to the atmosphere. From H to B the card shows a slight amount of back pressure work due to the expulsion of the products of combustion, but in general the pressure has fallen to that of the atmosphere at the end of the stroke B, when the cycle of operations again commences.

Bottom Loop or Weak Spring Diagrams.—In view of the wide range of pressure which usually occurs in the cylinder of a gas engine, the scale of the diagram is often so selected that 1in. vertical movement of the pencil corresponds to a pressure variation of 200lb. per square inch. Still higher scales are sometimes employed, and hence it will be evident that under these conditions the small variations of pressure which occur during the admission of the charge and the expulsion of the used gases will not be visible, the whole merging into one line practically indistinguishable from that denoting the pressure of the atmosphere. Generally, therefore, the loop formed by the admission and back pressure lines is negligible in comparison with the area of the upper portion of the card D E F H D (Fig. 96), which latter is more generally regarded as the actual indicator diagram. Nevertheless, the fact should be kept in view that the area of the bottom loop represents negative work, and that when it becomes a measurable quantity—as when a lower working pressure is used or the "fluid resistance" of the engine is increased—the area of this loop is to be deducted from the upper loop in order to obtain the net indicated work done in the complete cycle.

In order to obtain information upon the action of the valves, the sufficiency or otherwise of the areas of ports, piping, etc., and in general to ascertain the loss of work due to fluid friction, it is necessary to take "bottom loop" diagrams, a much weaker spring being used than is employed in taking a working card, in order to exhibit the small

pressure variations on a fairly large scale. Examples of such diagrams from a 400H.P. Crossley gas engine are given in Figs. 97 and 98. Fig. 97 is the diagram obtained during a firing cycle, while Fig. 98 shows the card which results from a "missed" firing stroke when the engine takes in air only.

The necessity for protecting the indicator spring from over-compression when taking these low-pressure cards has already been referred to (Part I., page 124), and it is here only necessary to explain that the horizontal lines M N in the cards shown represent the pressure at which the stop comes into play, and have no significance whatever as portions of the indicator cards.

The Exhaust and Admission Lines.—Referring to Fig. 97, N S shows the exhausting end of the explosion stroke, and

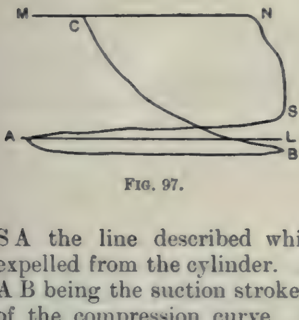

FIG. 97.

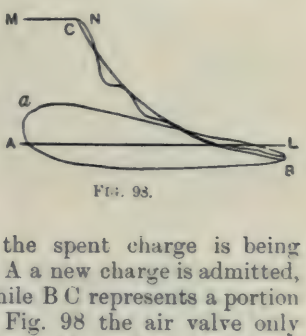

FIG. 98.

S A the line described while the spent charge is being expelled from the cylinder. At A a new charge is admitted, A B being the suction stroke, while B C represents a portion of the compression curve. In Fig. 98 the air valve only is opened (at a), and in consequence the resistance is somewhat increased, as indicated by the lower position taken by A B. The air charge is compressed along B C, and in the subsequent stroke expands along what is practically the same curve.

During the exhausting stroke the pressure rises as indicated by the back pressure line B a, this being largely due to the frictional resistance offered at the high piston speed (900ft. per minute) by the long exhaust pipe used in connection with the scavenging arrangement, and later, to the closing of the exhaust valve. At a the admission valve is opened, and the pressure falls rapidly in consequence.

As the area of the bottom loop in a gas engine diagram represents negative work, it is obviously desirable that the pressure during the charging stroke should approximate as nearly as possible to that of the atmosphere. Factors which determine how nearly this end can be attained are the regularity of the gas pressure, the piston speed, the length of the air pipe when one is used, and the design of the admission valve and port openings. Under normal conditions, and with "hit-or-miss" governing, the average fall of pressure during the suction stroke will usually not be less than 1lb. nor more than 2½lb. per square inch in engines up to 12 or 15B.H.P.; it may reach 3lb. in large engines. Excessive throttling during admission is shown in Fig. 99, the fall

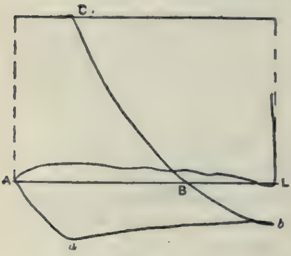

FIG. 99.

FIG. 100.

in pressure at *a* exceeding 6lb. per square inch, while at *b*, the end of the stroke, there is still a pressure deficiency of about 4lb. One result of this excessively low admission pressure is the greatly lessened volume of the charge admitted, for in place of a volume A L at the commencement of compression, the volume now available is that represented by A B only.

In engines governed by throttling the incoming gas and air, this effect is, of course, much intensified, especially when running unloaded. Fig. 100, which is a case in point, is from a Westinghouse gas engine when running light.

Late opening of the admission valve is shown in Fig. 101, the pressure falling as indicated by the line A B until the valve opens at B, when the pressure rises rapidly to the normal suction line.

Early closing of the admission valve will result in a fall of pressure as indicated at A B, Fig. 102, owing to the charge expanding, this portion of the line being retraced during the early part of the compression stroke.

Late closure of the admission valve is indicated in Fig. 103, the charge being forced back until the valve closes at A, and compression commences.

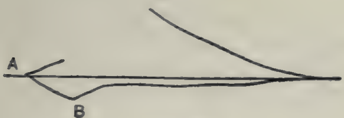

FIG. 101.

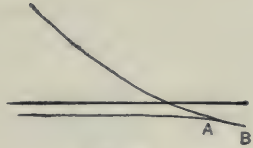

FIG. 102.

Late opening of the exhaust valve is shown much in the same way as in the steam engine (see Fig. 30, page 37).

Early closing of the exhaust is indicated by a somewhat rapid rise of pressure as at A B, Fig. 104. This defect is often due to wear of the cams and rollers or other parts of the valve gear.

Undulations in the exhaust line, such as occur in Fig. 105, are due to pulsations of the exhaust gases during their expulsion from the cylinder. It will be noticed that the

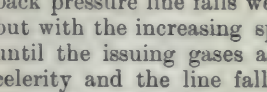

FIG. 103.

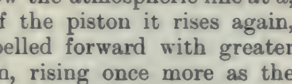

FIG. 104.

back pressure line falls well below the atmospheric line at *a*, but with the increasing speed of the piston it rises again, until the issuing gases are impelled forward with greater celerity and the line falls again, rising once more as the piston movement becomes slower near the end of the stroke.

Exhaust Silencers.—It is obvious that the use of exhaust silencers will somewhat increase the loss due to back pressure by reason of the resistance offered to the flow of the spent gases; while in extreme cases the retention in the cylinder of an undue amount of the products of com-

bustion will lead to after-firing in the exhaust pipe, and cause irregular running of the engine. But with a well-arranged silencer the loss due to increased back pressure should be immaterial, always provided that the silencer is regularly cleared of the deposit which will otherwise accumulate and increase the resistance.

Back-firing or explosions of a partial charge during the charging stroke are shown in the lower part of Fig. 106, the point *a* in the suction or charging stroke marking the point at which this explosion occurs, causing the line to rise suddenly as shown. The burnt gases are then compressed along L *b* and expand along *b c* L. Frequently the explosion occurs near the end of the suction stroke, as at *a*,

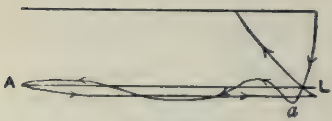

FIG. 105.

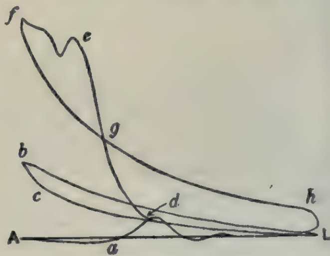

FIG. 106.

Fig. 107, where the rise in pressure is shown by *a b*; the subsequent compression is to *c*, and expansion to *d*, after which there follows a normal cycle as indicated by *f g h d*.

A frequent cause of back-firing is the admission of a charge weak in gas, which in consequence does not fire promptly, but burns, or rather smoulders, during the exhaust stroke. Hence, upon the admission of a further charge of gas during the succeeding suction stroke the mixture is ignited and back-firing results. In some engines back-firing may be attributed to the poor design of the combustion chamber, in which pockets are left which retain a small portion of the burning charge after the bulk has been expelled from the cylinder. Even the small recess formed by the indicator connection may suffice to retain a sufficient quantity of slow-burning gas to cause back-firing.

The Compression Line.—In general it may be said that the compression line of a gas or oil engine diagram shows a close approximation to that theoretically desirable. This results from the fact that during the compression stroke the charge of mixed air and gas is removed from all outside in-

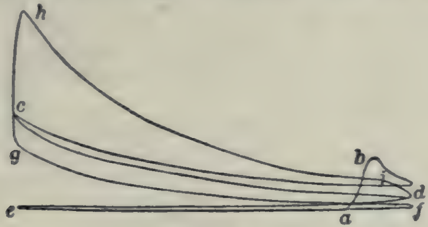

FIG. 107.

fluences, except in so far as the cooling by the jacket can be so considered. Hence practically the only element of uncertainty during this portion of the cycle is the possibility of pre-ignition.

By disconnecting the ignition and allowing the charge to be compressed and expanded, useful information can often be obtained as to the tightness or otherwise of the piston

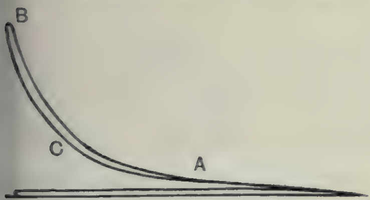

FIG. 108.

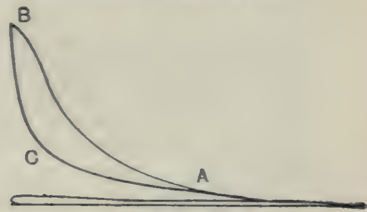

FIG. 109.

and valves. When these are in a satisfactory condition the compression card will appear as in Fig. 108, the charge being compressed as shown by A B and expanding again as shown by the slightly lower curve B C. Valve or piston leakage is indicated in Fig. 109, a considerable fall of pressure taking place near the end of the stroke when the compression pressure is greatest and the piston movement small.

Valve and piston leakage is frequently caused by a tarry deposit or oily residue which clogs the valve stems and piston packing rings.

It may be noted that the area of the card in Fig. 108 represents work lost due to the cooling of the charge in the cylinder. When the cylinder is not efficiently cooled, however, it is possible to obtain "cut-out" diagrams in which the expansion curve is above the compression curve, the resulting area representing work done by heat absorbed from the cylinder walls.

Pre-ignition.—The accidental firing of the charge during the compression stroke may be due to the failure of a charge to explode, thus resulting in the accumulation of an unduly rich charge, which is much more readily fired, during the next compression stroke. This effect is more marked in engines governed on the "hit-or-miss" system than in those governed by throttling the charge. A more frequent cause of pre-ignition, however, is the accumulation of deposits in the cylinder, partly from the lubricant used, and also owing to the accumulation of carbon from incomplete combustion, these deposits retaining sufficient heat to fire the charge prematurely. A slight projection of the asbestos used for joint-making may, under favourable conditions, become overheated and thus be sufficient to fire the charge, while among the structural defects which contribute to this end, inefficient jacketing is probably the most important.

Other causes of pre-ignition are over-compression, irregularity in the composition of the mixture in consecutive charges, an inefficient expulsion of the burnt gases from the cylinder during the exhaust stroke, and imperfect combustion.

A particularly bad case of pre-ignition is shown in the upper part of Fig. 106. Pre-ignition occurs at d, causing a sudden rise of pressure $d\,e$, opposing the motion of the piston, and finally the gases expand along $f\,h$. It may be noted in passing that of the several loops thus formed, both the areas $g\,e\,f\,g$ and $\mathrm{L}\,b\,c\,\mathrm{L}$, and also the small loops in the suction stroke below the atmospheric line, represent negative work done during the two cycles, the corresponding positive work areas being $g\,d\,\mathrm{L}\,h$, and the loops in the suction stroke above A L.

It will be readily understood that pre-ignition as early in the compression stroke as that indicated in Fig. 106 will result in shocks which are highly detrimental to the engine. When the premature firing of the charge takes place as late in the stroke as shown in Fig. 110, the opposition to the more slowly-moving piston, although less marked, will cause the engine to run very irregularly, while it is obvious that the full value of the charge is not by any means realised. The cause of the pre-ignition shown in Fig. 110 was over-heating of the exhaust valve—an effect often only noticed when the engine is working up to its full load and there are no cut-out strokes to assist in keeping the valve below the igniting temperature.

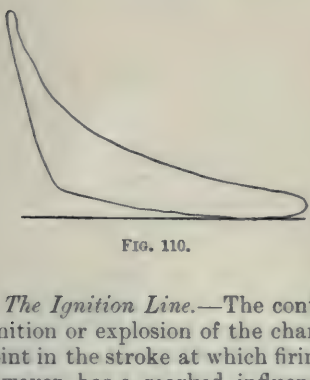

FIG. 110.

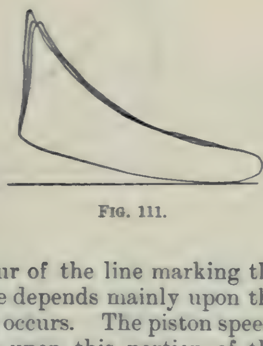

FIG. 111.

The Ignition Line.—The contour of the line marking the ignition or explosion of the charge depends mainly upon the point in the stroke at which firing occurs. The piston speed, however, has a marked influence upon this portion of the diagram, while the effectiveness with which the air and gas are mixed is also not without influence. Timing valves, which are usually fitted to all but small engines, ensure the firing of the charge at a definite point in the stroke, but by reason of the other factors mentioned, it frequently occurs that while the admission, compression, and exhaust lines in a consecutive series of diagrams may coincide exactly, a more or less marked divergence is seen in the corresponding ignition and expansion lines. Fig. 111, which is a diagram from an $8\frac{1}{2}$ by 16 in. gas engine fitted with a timing valve, and running, unloaded, at 180 revolutions per minute, shows this effect, but a more marked divergence is frequently met

with. Fig. 112 is a diagram from an engine without a timing valve, which, as may be expected, shows a wider diversity in the firing and expansion lines.

The inclination of the ignition line to the left, as in Fig. 110, or to the right, as in Fig. 111, is generally characterised as early and late firing respectively; while a vertical firing line is held to indicate that the explosion is effected on the dead centre. Of the three varieties, early firing must always be disadvantageous, and while firing on the dead centre is undoubtedly to be preferred from a thermodynamic point of view, there is no doubt that a slightly late firing leads to smoother running. Tests have been made which show clearly that within limits the frictional losses in a gas

FIG. 112.

engine diminish as the ignition is later, this result no doubt being due to the lessened direct thrust on the crankshaft bearings as compared with dead-centre firing. Nevertheless, in ordinary horizontal gas engines the firing is generally so timed as to give a vertical ignition line, and for this purpose the inlet valve may be set to open when the crank is about 15° below the centre. Naturally this angle will vary with the piston speed. In multiple-cylinder engines this practice would entail a risk of back-firing, and in such engines the inlet valves are usually set to open when the crank is some 15° above the centre.

The temperature and adjustment of the ignition tube have an all-important influence upon the rapidity with which the explosion is effected, and hence upon the shape of the ignition line. The "firing lead" of the timing valve and the piston speed are obvious factors; while it will be readily understood that the richness of the charge has also a con-

siderable effect upon the rapidity with which combustion takes place. The latter point is well shown by Fig. 113, in which the firing is so very late that eventually a charge entirely fails to explode. As a consequence, the next admission gives a total charge of double strength which readily fires on the centre, giving the single vertical firing shown, terminating in a horizontal line which marks the highest limiting position of the indicator piston. With very weak charges the distinction between the ignition and expansion lines entirely disappears, as shown by Fig. 114, the charge burning slowly throughout the entire stroke and leading to the back-firing previously referred to.

The three ignition lines shown in Fig. 115 are from the

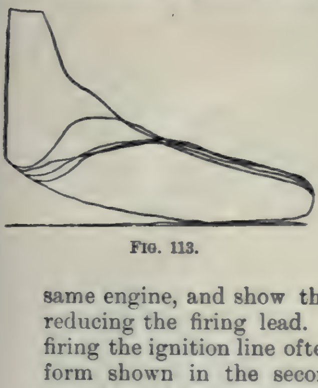

FIG. 113.

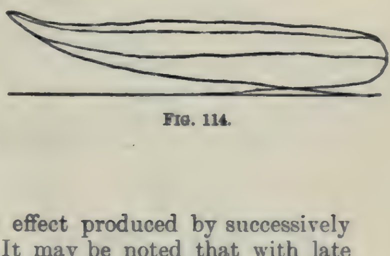

FIG. 114.

same engine, and show the effect produced by successively reducing the firing lead. It may be noted that with late firing the ignition line often appears of the concave or hollow form shown in the second example, which is to be distinguished from the *rounded* form of ignition line, indicating a weak charge. The abrupt break in the third ignition line is attributable to imperfect mixing of the air and gas.

The firing of the charge, taking place as it does at a point where the piston movement is small, is but imperfectly shown in the ordinary diagram, and hence it would appear that the crossed diagram, referred to on page 62, merits attention in this connection as offering a convenient means for more closely investigating the explosion of the charge and its attendant phenomena. An arrangement devised by H. Güldner * gives a diagram somewhat of this kind. In this

* "Zeitschrift des Vereines deutscher Ingenieure." vol. xliv., page 1730.

case only one complete oscillatory movement is given to the indicator drum instead of two, the indicator being operated from the half-speed shaft, and the reducing motion so arranged as to show firing occurring when the drum is at mid-stroke. Fig. 116 shows the form of the resulting

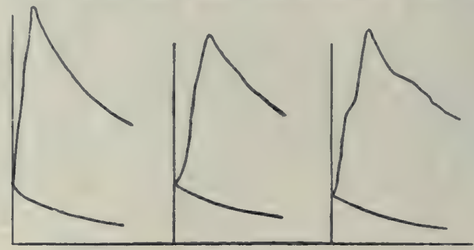

FIG. 115.

diagram, the dotted lines indicating the modified form which is now assumed by the apparently vertical ignition line. It would appear that the object in view could be equally well attained by driving the indicator drum by a cord attached to a pin set eccentrically in the end of the crankshaft, so as to give the required travel to the drum, and so placed in relation to the engine crank that the drum is at its mid-travel as the engine passes the inner dead centre.

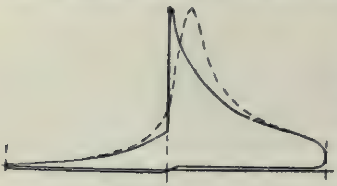

FIG. 116.

Körting has taken diagrams on a strip of paper running at about 2ft. per second, obtaining in this way pressure-time diagrams which are of considerable value as a means of examining ignition and combustion phenomena.

The Expansion Line.—The character of the expansion line of the gas engine diagram is largely influenced by the form of the ignition line, and hence, indirectly, is chiefly dependent upon the factors considered in the preceding section. With conditions favourable to rapid combustion, and a firing line vertical or nearly so, the point of maximum

pressure frequently coincides with the upper limit of the ignition line, and often in such cases the junction of the ignition and expansion lines forms a sharp peak. More generally, however, the explosion is not completed until the piston has moved forward to some extent, this resulting in the more usual formation of the rounded corner seen in Fig. 117. A flatter corner as in Fig. 118 is to be viewed with suspicion as pointing either to a defective indicator or to the spring reaching the limit of its compression before the full pressure is attained. The remedy in either case is obvious.

The subsequent expansion curve usually lies above the theoretical curve due to the true adiabatic expansion of

Fig. 119.

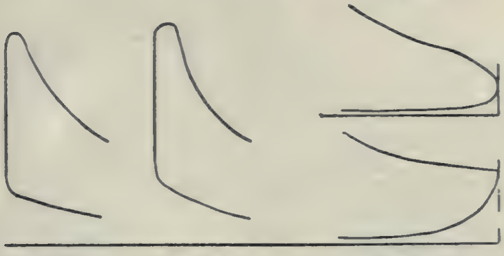

Fig. 117. Fig. 118. Fig. 120.

the charge, showing that heat is added to the gases during expansion in a water-cooled cylinder. To account for this effect various theories have been advanced. It appears reasonable to assume that combustion cannot be effected completely before expansion commences, in which case heat would be evolved during the early part of the expansion period. More elaborate theories have been advanced which imply physical changes in the working fluid, as "dissociation," increase of the specific heats of the expanding gases, etc. The "stratification of the charge"—a theory due to Otto, now finds little support; but at least one authority, Professor Witz, maintains that this addition of heat is to be fully accounted for by the action of the cylinder walls.

Occasionally undulations appear in the expansion line (other than those due to the indicator) which under

conditions favourable to their formation may produce waves of great amplitude. The effect appears to be due to a wave motion in the gases during combustion.

In the majority of cases the expansion period is not prolonged beyond from about 80 to 90 per cent. of the stroke. This gives a well-rounded corner to the toe of the diagram (Fig. 119), the slight loss of forward pressure area incurred being practically offset by the lessened back pressure area. Moreover, with early release the sudden expansion of the spent charge takes place mainly in the cylinder. On the other hand, when release does not occur until the end of the stroke (Fig. 120), the whole of the

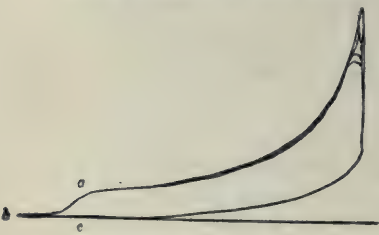

FIG. 121.

FIG. 122.

FIG. 123.

sudden expansion takes place in the exhaust pipe, causing a noisy discharge; where this is objectionable, its prevention by means of muffles or silencers involves a further slight increase in the back pressure work of the engine.

DIAGRAMS FROM TWO-CYCLE ENGINES.

Cards from two-stroke gas engines differ from those given by the more generally used four-stroke type, particularly in regard to the point of exhaust opening. Thus in Fig. 121, a diagram from a 600H.P. two-stroke engine, exhaust commences at the point *a* when about four-fifths of the expansion stroke has been completed. The pressure falls rapidly, and during the last eighth of the stroke the products of combustion escape at a pressure of some 10 or 12lb. in excess of that of the atmosphere. As in this case the spent gases are not expelled by the piston, the cylinder is cleared by a scavenging air flush. A fresh charge is now admitted to the

cylinder, and at the point *c* compression commences, the subsequent events of firing and expansion following exactly as in the four-stroke cycle.

It will be evident that with this method of working, the usual suction and back pressure lines forming the bottom loop no longer appear, and to determine the loss from fluid friction (which will be greater in this case), diagrams are required from the pump by which the charge is slightly compressed and forced into the cylinder.

In large engines, such as the Körting two-cycle type,

FIG. 124.

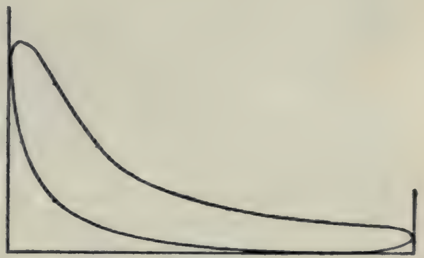

FIG. 125.

separate air and gas pumps are provided, the constituents of the charge being delivered under pressure, and mixing immediately in front of the admission valve. Representative cards from air and gas pumps are shown in Figs. 122 and 123.

OIL ENGINE DIAGRAMS.

In no essential particular does the diagram obtained from the oil engine differ from that given by the gas engine, and in general what has already been said in connection with the latter applies equally well to the former. Some slight modification of form arises from the different systems of vaporising the oil, and in most engines it will be found that the loss due to fluid friction is more pronounced than in the gas engine, a somewhat higher back pressure and the fall during the suction stroke frequently producing a very decided loop. In Fig. 124, which is a diagram from a Crossley oil engine, a pronounced

loop is seen even at the high scale of spring employed ; it is, however, mainly confined to the outer half of the stroke, and the average loss shown is less than that often met with in other types of oil engines. As may be anticipated, the mixture of the oil vapour and air is not so intimately effected as in the gas engine, this being often evidenced by slight irregularities in the outlines of both compression and expansion curves. Further, as a mixed charge of oil vapour and air is much more liable than a gas engine charge to ignite spontaneously during compression, the risk of pre-ignition imposes a much lower limit to the compression pressure; indeed, it is generally true that oil engine diagrams will show both a lower compression and a lower explosion pressure than are usually found in gas engine diagrams.

In the Diesel petroleum engine exceptionally high pressures, sometimes reaching 600 or 700lb. per square inch, are employed, but as air alone is admitted during the compression stroke, pre-ignition is impossible. Firing is effected by injecting oil into the compressed air, the temperature of the latter being sufficient to ignite the spray of finely-divided oil vapour, and hence to explode the charge. Under these conditions of working the combustion of the charge is practically perfect, and hence the heat efficiency shows a distinct advance upon other types of gas and oil engines. An important advantage is also obtained in the ease and precision with which the governing of the motor is effected, this being arranged for by by-passing more or less of the charge as it is delivered from the oil pump. An indicator diagram from a Diesel engine is shown in Fig. 125.

CHAPTER VI.

DIAGRAMS FROM AIR COMPRESSORS, PUMPS, ETC.

AIR COMPRESSOR DIAGRAMS.—Diagrams from air compressing machines exhibit a cycle of operations which may be regarded as the inverse of that indicated by the steam engine diagram. Thus, referring to the theoretical diagram, Fig. 126, the line L A represents the unimpeded flow of air into the cylinder during the whole of the piston stroke. At A the admission valve is

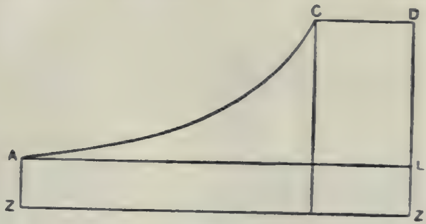

FIG. 126.

closed and the returning piston compresses the air as indicated by the compression curve A C, which latter, for the best results, should be isothermal in character; in other words, the curve should be a rectangular hyperbola. At C the delivery valve opens and the contents of the cylinder are discharged at a uniform pressure equal to that of the receiver, as indicated by the horizontal delivery line C D. At D the delivery valve closes, and simultaneously the admission valve reopens, resulting in the formation of the vertical D L, indicating an instant fall to the lower pressure.

In practice, the chief divergences from the ideal method of working are due (1) to the resistance offered by both inlet and discharge valves and ports; (2) to the generation of heat during compression, which modifies the character of the compression; and (3) to the effect of cylinder clearance. In Fig. 127, which shows the effect upon the diagram of the several influences cited, both the adiabatic and isothermal curves are drawn in order to afford a comparison with the actual curve of compression. From what has been said in a previous chapter (page 14) it will be understood that the adiabatic curve A F represents the curve along which compression would take place if no heat was lost by the working fluid; while A H is the curve which the compression would

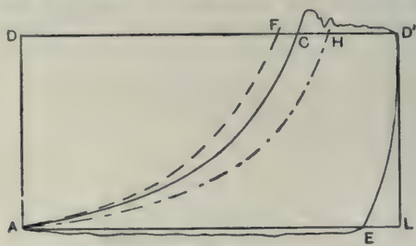

FIG. 127.

follow on the assumption that sufficient heat is abstracted to maintain the air at a uniform temperature. In practice, however, the compression curve is found to lie between the two reference curves, approximating to A F when little care is taken to cool the air, and more nearly approaching A H as more efficient means are provided for absorbing the heat generated. But even under the very best conditions the approach to isothermal conditions is not a close one. Designating the value of the curve by the equation $pv^n = C$, the limiting values of the index n are, $n = 1{\cdot}406$ for the adiabatic curve A F, and $n = 1$ for the isothermal curve A H. In practice, n may be as high as $1{\cdot}4$; but for ordinary single-cylinder compressors with a water jacket, n has often a value of $1{\cdot}3$. In the better class of wet compressors with spray injection, n often lies between $1{\cdot}25$ and $1{\cdot}20$; while for large compound compressors of good design, a value as low as

$n = 1{\cdot}15$ has been reached. But whatever the value of the curve, it should lie evenly in the space enclosed by the two standard reference curves, definite changes in its direction or abrupt breaks in its contour indicating leakage past the piston or valves. When the leakage is very pronounced, the compression curve will fall distinctly under the isothermal curve. The importance of securing the isothermal condition as far as practicable arises from the fact that in general the excess heat in the delivered air is dissipated by radiation before it is used, and hence the discharge volume represented by C D′ will have then contracted to H D′, involving a loss of compression work approximately represented by the area A C H A. From this will be seen the desirability not only of cooling the air as effectually as possible during compression, but also of using the compressed air at as high a temperature as convenient.

Upon referring to Fig. 127, it will be seen that owing to the resistance and weight of the delivery valve the compression curve arises above the line D D′, representing the receiver pressure, and also that the delivery line presents a series of undulations, the latter being due to oscillations of the valve under the action of the springs employed to secure prompt closing at the end of the stroke. When the receiver capacity is small, these undulations are often absent, and the delivery line rises somewhat, falling more abruptly as the end of the stroke is approached.

In the absence of any provision for minimising the effects of clearance, the high-pressure air in the clearance space expands as the return movement of the piston commences, forming the expansion line D′E. This reduces the volumetric efficiency of the compressor from L A to E A, but does not entail a loss of power beyond that due to the small amount of heat lost by the clearance air. It is, however, obviously desirable to reduce the clearance effect as far as possible, and various methods of accomplishing this end have been adopted. One of the most simple plans consists in providing small by-passes at each end of the cylinder, so arranged that as the piston nears the end of the stroke, communication is established between the clearance space on one side of the piston and the suction space on the other. In this way the greater part of the clearance air expands into the

suction space, and is usefully employed in raising the pressure therein.

Reverting again to Fig. 127, it will be seen that the suction line E A exhibits a somewhat undulating course, due to the oscillation of the admission valve on its seat, and to the resistance offered to the inflow of air. The area included between this line and that of atmospheric pressure A L, represents an amount of lost work which rarely averages less than 1lb. per square inch for the whole stroke, and is often much in excess of that amount. Owing to the slowing down of the piston near the end of the stroke, the atmospheric pressure is usually attained before the point A is reached. If, however, owing to badly-arranged valves, the pressure is still below A L when the compression stroke is commenced, a further loss of volumetric efficiency will follow, as some movement of the piston must take place before the pressure reaches that of the atmosphere and the compression proper commences.

Diagrams from Compound Air Compressors.—From what has been said as to the prejudicial effect of the heating of the air during compression, it will be seen that this loss will be greater as the degree of compression is higher. Hence for pressures above 60 or 70lb. per square inch it is customary to effect the compression in stages, passing the air, after compression in the first cylinder, through an inter-cooler in order to reduce it to the atmospheric temperature. Further compression in a second cylinder follows, while if the desired final pressure is from 300 to 1000lb. per square inch, a second cooler is used and a third compression stage adopted. Similarly, for pressures of from 1000 to 3000lb., four- or six-stage compressors are often used, the air being alternately compressed and cooled in the manner described.

Diagrams from compound air compressors are combined in a manner essentially analogous to that adopted in combining compound cards from steam engines; much useful information can be obtained from the combined cards, more particularly in regard to the efficiency of the inter-cooler. In the combined card shown in Fig. 128, A F is the adiabatic and A H the isothermal curve, while A N P S represents the complete actual compression curve. The low-pressure card is L A N R, and the high pressure R P S D, while L A and

L M represent the comparative volumes of the two cylinders. From the diagram it will be seen that the action of the inter-cooler was quite complete, the volume R N discharged from the L.P. cylinder being cooled to atmospheric temperature, as shown by the H.P. compression commencing at the point P in the isothermal curve. Had the cooler action been less perfect, the air in the L.P. cylinder retaining a portion of the heat of compression would have been of correspondingly larger volume, and hence would have necessitated a higher pressure in the L.P. cylinder and involved a loss of work. By continuing the compression line A N up to C, it will be seen that an amount of work approximately

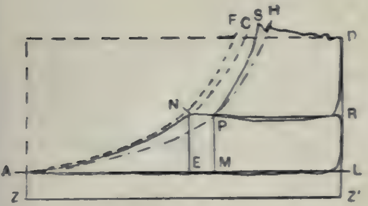

FIG. 128.

FIG. 129.

represented by the area N C S P has been saved by the inter-cooling of the air.

The diagram from a three-stage compressor given in Fig. 129 shows how, by increasing the number of stages, a still closer approximation to isothermal compression is attained. In this case the intermediate cooling effect is sufficient to bring the starting point of each compression line upon the isothermal curve. Hence the lost work is represented by the three triangular areas formed by the compression lines overlapping the isothermal curve, to which has to be added the small loop-like areas formed by the overlapping of the several cards, the amount of which depends upon the resistance of the passages between the cylinders and inter-coolers.

Blowing Engine Diagrams are a variety of air-compressor cards in which the chief points to be noted are the

losses due to clearance and to the resistance of the valves. Although the pressures dealt with are low, rarely exceeding 25lb. per square inch, the volumes and piston areas are large, and hence in these engines the loss above referred to becomes increasingly important. Fig. 130 is a card from a blowing engine with unduly large clearance spaces, involving a loss of about nine per cent. of capacity. The blowing engine diagram Fig. 131 shows the extent to which the loss due to valve resistance often occurs in these engines, the mean loss during admission being 10·5 per cent., and during discharge 12·1 per cent. of the power usefully expended.

Diagrams from Ammonia Compression Machines differ but little from those considered in the foregoing sections, except with regard to the suction pressure, which is above that of

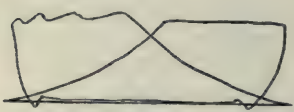

Fig. 130.

Fig. 131.

the atmosphere; hence the resistance offered by the inlet valves cannot well be ascertained by comparing the suction with the atmospheric lines, as in air compressors. However, gauges are almost invariably fixed to both the suction and delivery pipes, and by comparing the pressures thus indicated with those shown by the indicator diagram, the work due to valve resistance may be readily determined, always provided that the indicator (which for ammonia work must be of special construction—see Part I., page 57) has been found, by testing, to be in agreement with the pressure gauges employed.

Various devices have been adopted with the object of minimising the loss due to the expansion of the gas contained in the clearance spaces, and in one form of vertical compressor these spaces are filled with oil which is maintained in constant circulation. This method has the advantage not only of nullifying the clearance spaces, but also of assisting in the cooling of the compressed gas.

Fig. 132 is a diagram taken from a 14in. compressor of this type running without oil, while Fig. 133 is the corresponding card taken when the circulation of the oil has been restored. In each case E H is the isothermal and E F the adiabatic

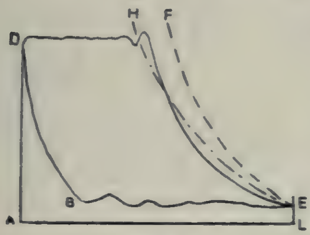

FIG. 132.

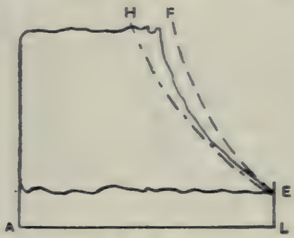

FIG. 133.

curve. Leakage past the piston is clearly indicated in Fig. 132 by the falling of the compression curve below E H, confirmation of this view being found in the form of the expansion curve D B. A comparison with Fig. 133 shows the marked gain in volumetric efficiency which results

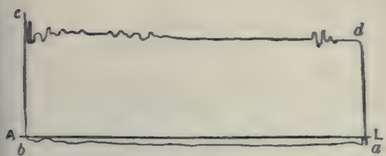

FIG. 134.

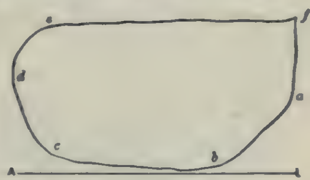

FIG. 135.

from the use of the clearance oil, and also an improved, but by no means perfect, compression curve.

Pump Diagrams.—The theoretical diagram from a water pump is a plain rectangle—a figure very closely approximated to in cards from water-works pumps, of which Fig. 134 is a good example. Commencing at *a*, the suction stroke is performed at an average pressure of about 3lb. per square inch below that of the atmosphere. At *b* suction ceases, and with the reversal of the plunger the pressure rises immediately to *c*, the peaks formed being

mainly due to hydraulic shock in the indicator. The subsequent minor fluctuations are caused by slight changes of pressure within the pump, those near the end of the delivery stroke being due to the influx of water into the main from another of the three cylinders of the pump. The pressure against which these pumps operate is 89lb. per square inch; the plungers are 33in. in diameter and have a stroke of 60in. The card shown in Fig. 134 was taken at 15 revolutions per minute.

Leakage past the plungers of water pumps, or through the valves, is indicated by a rounding of the corners of the diagram as in Fig. 135, which is a card taken from a 4in-boiler feed pump. Leakage of the delivery valve is prob-

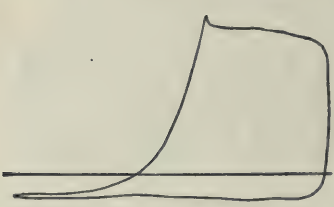

FIG. 136.

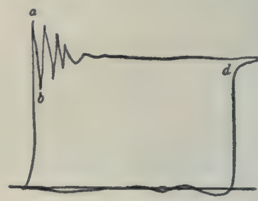

FIG. 137.

ably the cause of the marked curving at the commencement of the suction line *a b*, the effect becoming again evident as the plunger slows down towards the end of the stroke, causing curving of the line as at *c d*. Leakage past the piston or of the suction valve would account for the rounding at *d e*.

When feed and similar pumps are run much above their intended speed, the resistance to the inflow of water considerably reduces the amount drawn in at each stroke. Hence not only does the suction line appear well below the atmospheric line, as in Fig. 136, but also, as there shown, a large portion of the stroke is completed before the pressure rises to that of delivery. A somewhat similar effect is seen in cards from feed pumps drawing water from a hot well, in which case the barrel is partially filled with vapour and air, and the capacity of the pump is much reduced.

Sluggish action of the delivery valve, by delaying the closing to pressure, will result in the formation of a toe or beak as at *c*, Fig. 137, where a portion *c d* of the return stroke is completed before the valve is completely closed and the pressure falls to the point which marks the true commencement of the suction stroke. The more or less violent fluctuations of pressure, as at *a b*, which often occur during the early portion of the delivery stroke, are due to hydraulic shock, the effect being, as already mentioned, partly due to the indicator. When air vessels of a suitable volume are provided, a much smoother action is obtained, with still better results if such chambers are fitted on the suction as well as on the delivery side of the pump.

Diagrams taken from the water side of duplex steam pumps at various speeds, while presenting no features not

FIG. 138.

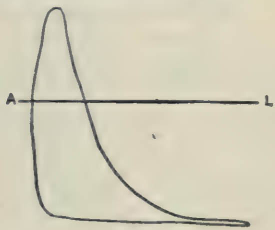

FIG. 139

already alluded to, will frequently be found to differ materially in their lengths, the cards obtained at the low speeds being shortened by reason of the lower velocity of the moving parts and their consequent earlier stoppage in the stroke.

Air Pump Diagrams, under the best conditions of working, often approximate in form to those obtained from air compressors. This is more especially the case with single-acting pumps working in connection with surface condensers, as the quantity of air and water to be dealt with is considerably less than with jet condensers. Thus in Fig. 138, which is a diagram from a vertical single-acting air pump working in connection with a surface condenser, the maximum discharge pressure is 2lb. per square inch above the atmosphere. In Fig. 139, however, which is a card from a single-acting

air pump used with a jet condenser, the discharge pressure attains a maximum of 10lb. per square inch.

When the valves of an air pump are allowed to become foul owing to deposits of grease, etc., they will often fail to

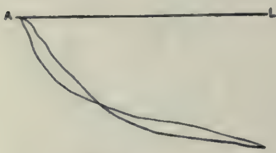

FIG. 140.

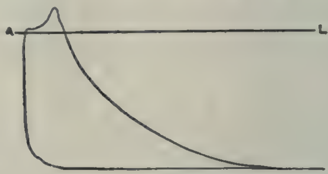

FIG. 141.

lift, and for one or more strokes the air and water contained in the pump will be alternately compressed and expanded, until finally the accumulation of water in the condenser causes the valves to open, and the pump fulfils its intended function. A diagram resembling that shown in Fig. 140 is obtained

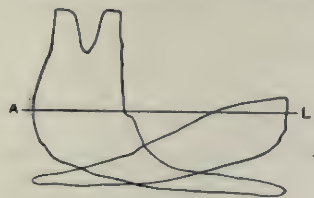

FIG. 142

during such an idle stroke, Fig. 141 being the corresponding diagram obtained under normal conditions of working.

Diagrams from a vertical double-acting air pump of the bucket-and-plunger type are shown in Fig. 142. The irregularity in the discharge line from the larger diagram is frequently met with, although usually more than one undulation is found. Such fluctuations of pressure are doubtless due to the oscillations of the valve and the spasmodic expulsion of the air and water.

CHAPTER VII.

DIAGRAM CALCULATIONS.

INDICATED HORSE-POWER.—The indicator diagram furnishing as it does a graphic record of the pressure acting upon the piston throughout the stroke, finds an important use in connection with the measurement of engine power. For this purpose it becomes necessary to deduce from the diagram the *mean effective pressure* exerted during the stroke; in other words, to determine that pressure which, *acting uniformly* throughout the stroke, would produce a rectangular diagram precisely equivalent in area to the actual card. Then if P is the mean effective pressure in pounds per square inch, A the area of the piston in square inches, L the length of the stroke in feet, N the number of *strokes* (= *twice* the revolutions) per minute, and I.H.P. the indicated horse-power,

$$\text{I.H.P.} = \frac{P\,L\,A\,N}{33{,}000} \quad\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

When the cylinder diameter D in inches is given,

$$\text{I.H.P.} = \frac{P\,L\,D^2\,N}{42{,}017} = 0{\cdot}0000238\ P\,L\,D^2\,N \quad\ldots\ldots \quad (2)$$

If the piston speed S in feet per minute is given, then

$$\text{I.H.P.} = \frac{P\,D^2\,S}{42{,}017} = 0{\cdot}0000238\ P\,D^2\,S \quad\ldots\ldots\ldots \quad (3)$$

It should be noted that the net or *effective* piston area is required; hence in ordinary cases allowance should be made for the piston rod by deducting one-half its area from A in (1). If a tailrod is used, one-half the sum of the areas of the two rods should be deducted. The piston rod can be

allowed for in (2) or (3) by deducting one-half the square of the rod diameter if a single rod is used, or one-half the sum of the squares of the two rod diameters if a tailrod is used.

In four-cycle gas and oil engines no such allowance is ordinarily required; but in some forms of two-cycle gas engines the correction will be necessary. It is to be observed that in this case N in (1) is always to be the number of *firing* strokes per minute.

When a number of cards from the same engine are to be dealt with, the horse-power calculation may be simplified considerably by expressing the non-variable factors of the equation, in the form of a constant. Thus, for any engine, the area of the piston (A) and the length of stroke (L) are

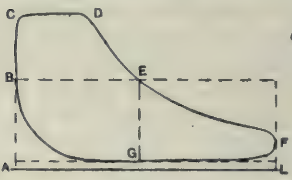

FIG. 143.

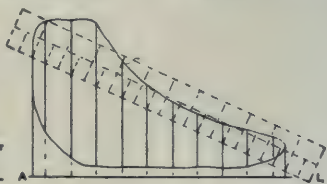

FIG. 144.

unalterable factors, and hence matters may be simplified by multiplying out, once for all, the value of the "engine constant," $\frac{A \times L \times 2}{33{,}000}$. Then this constant multiplied by the number of revolutions per minute, and by the mean effective pressure (P) deduced from the diagram, will give the I.H.P. developed.

A still further simplification may be effected when the piston speed remains uniform, by multiplying the constant found as above by the number of revolutions per minute, this giving a "piston constant" which, when multiplied by the mean effective pressure, gives the I.H.P. as before.

Mean-Pressure Calculations.—It may be said that the area A C D F L A under the forward pressure line of the diagram shown in Fig. 143 represents work done by the steam *on* the piston; while the area A B G F L A under the back pres-

sure line represents work done *on* the steam *by* the piston. The difference is the effective work area B C D F G B, and for the present purpose it is quite immaterial that the back pressure work area is really coincident with the forward pressure area of the card from the other end of the cylinder, since obviously this can have no bearing upon the result, as both the cards are ultimately taken into account.

Except in the case of looped diagrams (page 117), it is unnecessary to separately compute the forward and back pressure areas as just described, the required M.E.P. being obtained more directly and accurately by dividing the area of the actual diagram as B C D F G B by its length as A L. The result gives in effect the height of the mean-pressure

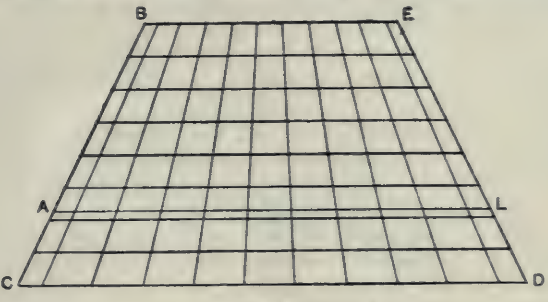

FIG. 145.

ordinate as G E, and this, multiplied by the scale of the spring employed, gives the required mean pressure.

A more direct method of finding the M.E.P. consists in dividing the diagram in the manner shown in Fig. 144. The object here is to obtain approximately the mean height of the 10 rectangular sections into which the diagram may be assumed to be divided; but, as will be seen, this resolves itself into drawing 10 equally-spaced ordinates, so that the spaces remaining at each end are each one-half of the common interval, as shown in the illustration.

A convenient method of effecting the required division, by the aid of an ordinary rule or scale, is indicated in the figure; but for continued use it is preferable to make a special scale divided in the particular matter required.

Another method of locating the ordinates is by the aid of the diagram shown in Fig. 145, which may be drawn on stout card, C D being conveniently made about 6in. long, B E, 2in., and the vertical height about 3in. In using the diagram, the exact length of the card to be divided, as A L, is marked off on the edge of a strip of paper, and this is placed horizontally on the diagram, and in such a position that A and L fall upon B C and E D, as shown. The intermediate divisions are then marked off on the paper strip, transferred to the card, and the ordinates drawn in. The author prefers to draw the diagram upon stout tracing paper or cloth, and to apply this directly to the card, locating the ordinates by pricking through the division lines where they cross the card length line A L.

Special parallel rulers are sometimes supplied for spacing the ordinates, but as these usually divide the length into 10 *equal* parts they are not so convenient as could be wished.

Each of the ordinates in Fig. 144 may be measured by the scale corresponding to the spring used, and the pressures so obtained added together; the result, divided by 10, will give approximately the M.E.P. per square inch throughout the stroke. A more accurate and convenient method consists in mechanically adding the ordinates together by means of a strip of paper or card having a straight edge. A mark near one end of this strip serves as a starting point from which the lengths of the several ordinates are stepped off in succession, so that the total length of the marked portion of the strip is equal to the sum of the 10 ordinates. This length, divided by 10 and multiplied by the scale of the spring, gives the M.E.P. required.

Special instruments for effecting the summation of the ordinates with greater accuracy have been introduced from time to time Of these the *Meanometer* consists of a rule bearing scales and fitted with a sliding runner or cursor. Hale's *Slide Scale* comprises a suitably divided scale with a sliding vernier, by which readings to 0·001in. can be made. In using this instrument, the ordinates are drawn as described, and the zero on the slide having been brought opposite to the zero on the scale, the scale is placed on the first ordinate with the slide on the lower line of the diagram. The slide

is then moved upward along the ordinate until the edge of the slide cuts the upper line of the diagram. When this operation has been repeated for each of the 10 ordinates, the reading on the slide is taken, and this, multiplied by the scale of the spring used, gives the M.E.P. in pounds per square inch.

Among other simple devices for the same purpose, mention may be made of the *White-Bean Area Scale*, which consists of a sheet of transparent celluloid on which is printed a series of equidistant lines, alternately full and dotted. The diagram is placed under the celluloid so that each extreme end of the figure lies equidistant between a pair of dotted and solid lines. The lengths of the several ordinates lying within the outline of the diagram are then successively

FIG. 146.

marked off on a strip of paper, and the total length measured on one or other of two scales (giving sq. in. or sq. cm.) marked on the upper part of the celluloid sheet. This area, divided by the length, gives the M.E.P.

Low's *Diagram Measurer* is of somewhat similar construction, but transverse scale lines are added. By sliding the celluloid sheet over the diagram (against a straight-edge) the sum of the intercepted ordinates can be read off on a scale at one side, giving the area of the figure in square inches.

The plan sometimes followed of measuring separately those portions of the ordinates above and those below the atmospheric line, and adding the results, has nothing to recommend its adoption, but rather the reverse, since it increases errors of measurement and is wasteful of time.

When vibrations of the pencil distort the diagram to any marked degree the ordinates should be measured to a line drawn midway between the crests and hollows of the undulations.

Diagrams which include one or more loops are best dealt with by the indirect method described in connection with Fig. 143. Thus, in such a case as shown in Fig. 146, the card should be divided and ordinates drawn as before. Then the sum of the ordinate lengths between A L and the forward pressure line B G F E is taken, and from this is deducted the sum of the ordinate lengths between A L and the back pressure line E D C B. The difference divided by 10 and multiplied by the scale of the spring will give

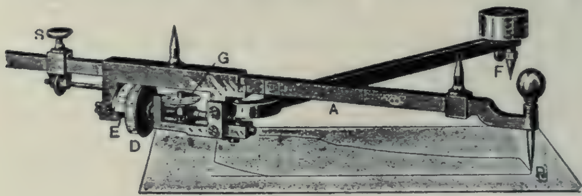

FIG. 147.

the M.E.P. required, which is in effect that due to the subtraction of the two shaded areas of negative work from the unshaded diagram area.

In diagrams from four-cycle gas engines the (negative) area of the loop formed by the suction and exhaust lines is usually a negligible quantity. In two-cycle engines the work done in the air and gas pumps is more readily calculated. It should be deducted from the positive work card to obtain the true indicated horse-power of the engine.

Planimeters.—A more accurate and expeditious method of obtaining the area of an indicator card—and hence the required M.E.P.—is by the use of some form of mechanical integrator or planimeter. Various types of this intrument are made, but the modified form of Amsler's Polar Planimeter shown in Fig. 147 is found very convenient for the present purpose, as it enables the M.E.P. to be read directly without first ascertaining the area of the card and dividing this by

its length. As will be seen, the instrument consists of two metal arms hinged together, the needle point F at the free extremity of one of the arms providing a pivot, while the other arm A is furnished with a tracing point P with which to trace the outline of the diagram to be measured. The roller wheel D upon which the instrument moves is provided with a scale graduated to record the area traced, the reading being facilitated by the addition of the vernier E and the counting disc G.

In using the planimeter it should be placed relatively to the diagram, somewhat in the position shown in Fig. 148, so

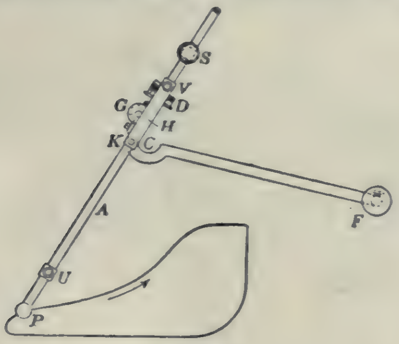

FIG. 148.

as to allow perfect freedom of movement in tracing the outline. The weight F is then placed in position, and the needle point pressed into the board carrying the instrument, and upon which the indicator card is fixed. It is desirable also to mount upon the board a piece of flat unglazed cardboard upon which the roller wheel can freely travel. Placing the pointer P at any convenient point upon the outline of the figure to be measured, the measuring wheel is to be either adjusted to zero or the present reading taken. Then the tracing point is carefully moved round the figure, in the direction of the hands of a watch, until the starting point is again reached. The reading given (either by direct observation or by subtracting the old reading from the new, according to the method pursued) will under ordinary

conditions be that of the area of the diagram in square inches.

The illustration of the recording mechanism of the planimeter given in Fig. 149 will serve to explain the manner of reading the vernier scale E. Referring to the scale on the wheel D, it will be seen that each of the 10 figured spaces is subdivided into 10 equal parts. The vernier scale is also divided equally into 10 parts, but the spaces are each $\frac{1}{10}$th less than those on D; in other words, 10 divisions on E are equal to 9 divisions on D. As shown in the illustration, the instrument is arranged to give the *area* of a figure, one complete revolution of D corresponding to 10sq. in. The number

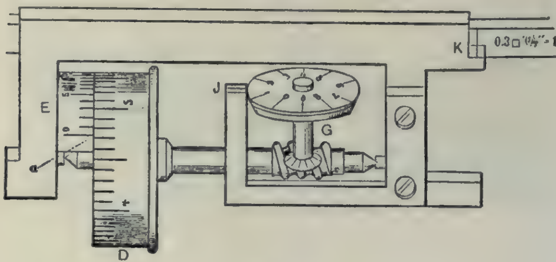

FIG. 149.

of completed revolutions of D are recorded on the disc G, while the graduations on D are read against the zero of the vernier scale. Thus, the reading here shown consists of 1 ten, 4 units, 7 tenths, or 14·7, *plus* a fraction of a space on D, which is determined by following along the vernier scale and noting the number of the division line thereon which agrees with *any* division on the main scale. In this case it is seen that the third division on E is that sought, and hence the complete reading will be 14·73 sq. in. For as the spaces on E are each $\frac{1}{10}$th less than those on D, it is obvious that three spaces on E will be $\frac{3}{10}$ths less than three spaces on D. Hence, taking the coinciding graduations as a common starting point, it is clear that the small interval marked *a* will be $\frac{3}{10}$ths of a space on D, or three-hundredths of a square inch.

With the particular instrument shown in Fig. 147, however, the result can be obtained more directly in the form of the average or mean-pressure ordinate of the diagram. This is effected by loosening the screw S and so adjusting the bar A (Fig. 147) that the two steel points exactly contain between them the length of the diagram to be measured (Fig. 150). When adjusted in this manner, the figures on the counting disc G are to be read as hundreds, those on the roller wheel as tens, the intermediate graduations units, and the vernier reading tenths. The instrument is used in the manner explained, but it is desirable to commence with the wheel at zero. As, however, the reading obtained will be the average height of the diagram in *fortieths of an*

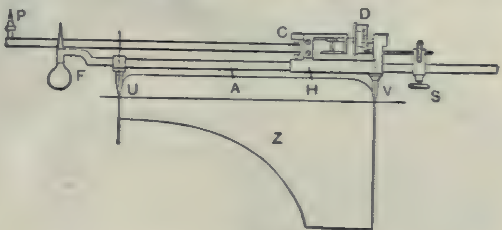

FIG. 150.

inch, the figures on the counting disc do not usually enter into the result. When the scale of the diagram is 40, the reading will be the required M.E.P. per square inch; for other scales it is necessary to divide the observed reading by 40 and to multiply by the scale of the diagram. Thus, if the roller wheel gives 42 and the vernier 8, the reading $= 42\cdot8 \div 40 = 1\cdot07$, and multiplied by, say 60, as the scale of the diagram, the M.E.P. is $1\cdot07 \times 60 = 64\cdot2$lb. per square inch. The same result can be obtained directly by multiplying the reading by a factor corresponding to the scale used. The factors corresponding to the springs most generally used are as follows:—

Springs...	8	12	16	20	24	30	40
Factors...	0·2	0·3	0·4	0·5	0·6	0·75	1·0
Springs...	50	60	80	100	120	150	180
Factors...	1·25	1·5	2·0	2·5	3·0	3·75	4·5

Ott's Compensating Planimeter, shown in Fig. 151, has a wide tracing range, and the accuracy of every reading can be immediately checked by a compensating measurement, and any error rectified.

The polar arm P and the tracer arm F are two independent parts, fitted separately in the case, but linked together for use by inserting the ball pin G into a socket in the roller frame. After the diagram has been traced and the reading taken, the pin G is lifted and the arms P and F swing round without moving the pole *p*. The pin G is then

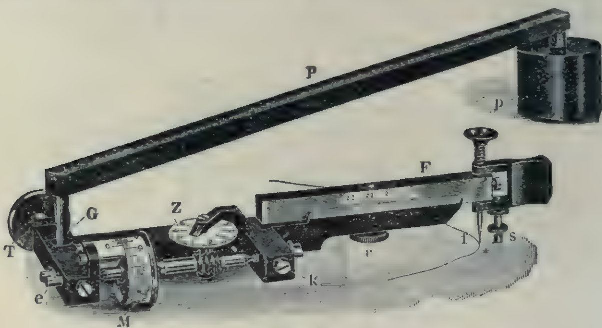

FIG. 151.

re-inserted in a position symmetrically opposite to its former position. The measuring operation is then repeated and the mean of the two readings obtained is the correct area of the figure.

To find the mean height of an indicator diagram the area obtained is divided by the length of the diagram measured between the extreme verticals. Assuming the area obtained is 4·27sq. in., whilst the extreme length of the diagram is 3·78in. ; then the mean height is $\frac{4 \cdot 27}{3 \cdot 78}$ or 1·13in. With the aid of a slide rule this division and the subsequent multiplication by the scale of the spring can be effected in a very convenient manner, while the results obtained are very accurate.

To obtain accurate readings with any planimeter the instrument must be clean and the base board perfectly flat.

The Coffin Averaging Instrument, shown in Fig. 152, is a very convenient modification of the planimeter, giving the M.E.P. directly without calculation or adjustment of

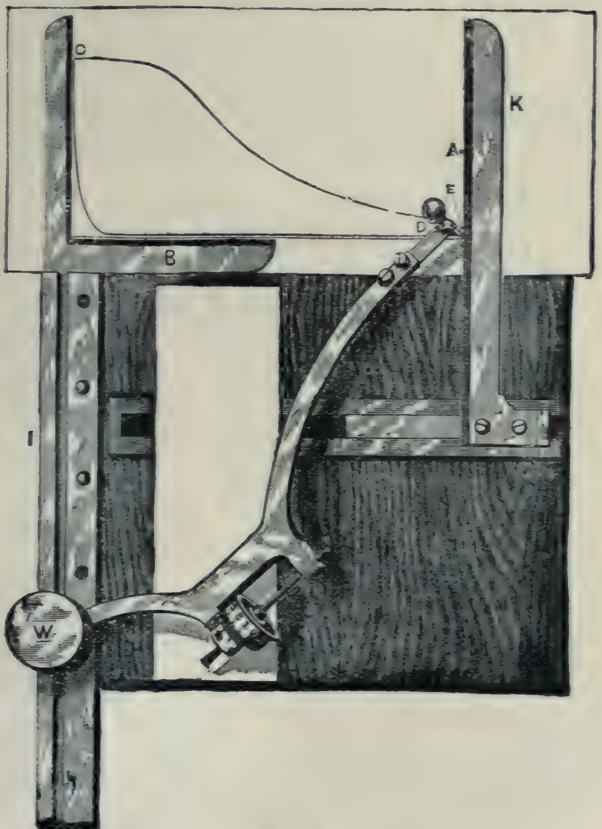

FIG. 152.

any kind. As will be seen, the board on which the apparatus is mounted carries a grooved metal plate I, being held in the position shown by a thumb-screw on the underside.

The indicator card is placed under the clips C and K, being adjusted so that the atmospheric line is near to and parallel with the lower arm B of the fixed clip, while the extreme left-hand portion of the diagram is placed close to the perpendicular arm of the same clip. The movable clip K—the lower end of which slides with a close fit in a lateral groove—is then moved along until the bevelled edge almost touches the extreme right-hand end of the diagram, the space left between the ends of the diagram and the corresponding clips being equal to one-half the diameter of the tracing point D.

The beam of the instrument is then placed on the board with the pin at the lower end resting in the groove in the plate I, and the weight W placed on the top of the pin to keep it securely in position. The tracing point is then moved to the extreme right hand of the diagram, and a slight indentation made in the paper at E, the point where the clip K and the diagram touch each other. The graduated wheel is next turned, so that its zero mark coincides with that on the vernier. The tracing point is now moved carefully along the line of the diagram in the direction of the hands of a watch until the circuit is completed. Keeping the eye upon the wheel, the tracer is then moved upward by sliding it along the edge of the clip K until the reading on the wheel again returns to zero, when another slight indentation of the paper is made, as at A. The beam of the instrument may now be removed, the clip K drawn away, and the distance E A, between the two indentations, when measured by the scale corresponding to the spring used, gives the M.E.P required.

If the area of the diagram is required instead of the average pressure, this can be obtained by taking the reading of the graduated wheel when the circuit of the diagram has been completed. The wheel is divided into fifteen principal divisions, each of which is again subdivided into five parts. A vernier scale attached enables the result to be read to 0·02 of a square inch.

The Hatchet Planimeter is an exceedingly simple appliance, which may be used either to obtain the area or the mean height of an indicator card. It consists essentially of a rod or wire of which the ends are turned down at right

angles, as shown in Fig. 153. The end A forms the tracing point, while B is flattened and brought to a knife-edge, but with a rounded or hatchet-shaped contour. In the special form used to obtain the M.E.P. direct from the diagram, the limb B is adjustable on the horizontal bar, and is so set that the distance between the centres of A and B is equal to the horizontal length of the diagram to be measured. As, however, the accuracy of the instrument is increased as A B is made greater, many prefer to use a bar of fixed length (10in. being convenient), and, after

FIG. 153.

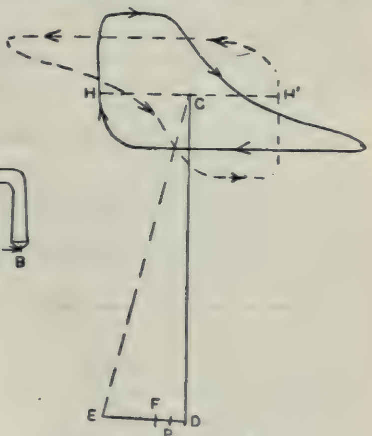

FIG. 154.

obtaining the area, to divide this result by the length of the card to obtain M.E.P.

In using the instrument, the pointer is placed approximately at the centre of gravity of the diagram C (Fig. 154), and the hatchet end D pressed into the paper or card upon which it rests. From C the pointer is moved in a direction at about right angles to C D until the point H is reached, the diagram being then traced over in the direction of the hands of a watch. When H is again reached the pointer is taken along H C, and when at the latter point another indentation E is made by pressing on the hatchet end of the instrument. Next, without disturbing the bar or the marks D and E, the

diagram is turned through 180° and the tracing process repeated, first moving to H′ and completing the circuit as before, but in the opposite direction. When C is again reached a final mark F is obtained, and midway between F and D the point P is found, giving E P as the result sought. If the instrument has been adjusted so that C D is equal to the diagram length, then E P, measured by the scale of the diagram, will give the required M.E.P. in pounds per square inch. If, however, the arm is of any other length (x), then E P is to be multiplied by x (giving the area in square inches) and divided by the diagram length to obtain the M.E.P. It will be seen that by using a length of 10in. the multiplication is readily effected by moving the decimal point.

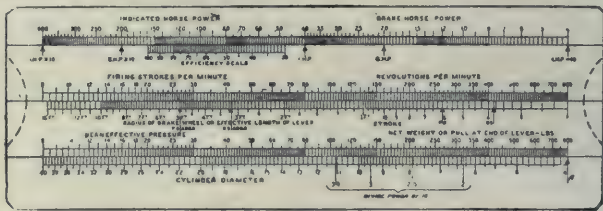

FIG. 155.

Horse-power Calculations.—The M.E.P. having been obtained by any of the methods described, the indicated horse-power can be calculated by the rules given on page 113, the several pairs of cards from multi-cylinder engines being treated separately and the results added to obtain the total power developed. These calculations, especially if numerous, are tedious and time-taking, and hence considerable advantage is derived from the use of specially-arranged slide rules, such as shown in Fig. 155, which represents, on a scale of about half full size, the author's Power Computer for Steam, Gas, and Oil Engines. This, as will be seen, consists of a stock on the lower portion of which is a scale of cylinder diameters, while the upper portion carries a scale of horse-powers. In the groove between these scales, two slides, also carrying scales, are so arranged as to be capable of sliding in edge contact with the stock and with each other.

In determining the *indicated horse-power* of gas or oil engines, the M.E.P. in pounds per square inch is set to coincide with the cylinder diameter on the lower scale, after which, the number of firing strokes per minute being set to the length of stroke, the indicated horse-power is read over arrow "I.H.P." In the case of steam engines, the number of firing strokes is, of course, the number of working strokes. If in the latter calculation the piston speed is given in place of stroke length and number of strokes, the given piston speed is set to 1ft. on the "stroke" scale, and the result read off as before. Other calculations may also be readily effected. Thus, the dimensions of an engine to develop a given power may be found by setting the "I.H.P." arrow to the required indicated horse-power, and the length of stroke to the number of strokes per minute, when under the M.E.P. is found the required cylinder diameter.

To find the *brake horse-power* of an engine by this instrument, the net weight in pounds acting on the brake strap (or the net weight at the end of the brake lever) is set to the arrow "A." Then setting the revolutions per minute to the effective radius of the brake wheel (or length of lever), the brake horse-power is read on the upper scale over the arrow "B.H.P." The calculation of the mechanical efficiency of engines, piston speed, ratio of compound engine cylinders, the delivery of pumps with any efficiency, the horse-power of belting, velocity ratio of pulleys and gear wheels, and the velocity of belts and ropes, are among the other principal purposes for which the computer may be advantageously employed.

Water Consumption Calculations.—Owing to cylinder condensation and other sources of internal waste in the steam engine, a much larger quantity of steam passes through the engine cylinder than is accounted for by the indicator diagram. Nevertheless, the calculation of the water consumption from the indicator card often supplies some interesting data on the comparative performances of engines of different types and under varying conditions of working, more particularly in regard to the relative economy of varying degrees of expansion.

If A is the area of the piston in square inches, L, the length of stroke in feet, N the number of strokes per minute, and

P the M.E.P. in pounds per square inch, then the volume displaced by the piston, or the *piston displacement*, will be

$$\frac{A}{144} \, L \, N \times 60 \text{cub. ft. per hour} \ldots\ldots\ldots\ldots\ldots \quad (4)$$

Hence the piston displacement per I.H.P. per hour will be

$$\frac{A}{144} \, L \, N \, 60 \div \frac{P \, L \, A \, N}{33{,}000} = \frac{13{,}750}{P} \text{cub. ft.} \ldots\ldots\ldots\ldots \quad (5)$$

From this it results that with a cylinder having no clearance, and hence no compression, the theoretical steam consumption per I.H.P. per hour would be found by dividing

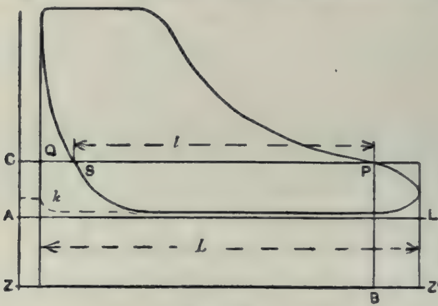

FIG. 156.

the constant 13,750 by the M.E.P. obtained from the diagram. In actual diagrams, with well-defined compression lines, the volume of steam retained in the clearance space may be readily allowed for by the following simple expedient. Through a point on the expansion curve as P (Fig. 156) taken just before opening to exhaust occurs, a horizontal line is drawn cutting the compression curve in S. Then if l is the intercepted length, L the whole stroke, and r the ratio of these lengths, $\frac{l}{L} = r$, and $l = r\,L$. Substituting $r\,L$ for L in the expression for piston displacement (4), it is seen that the piston displacement in cubic feet per I.H.P. per hour will now be $13{,}750 \, \frac{r}{P}$, and if W is the weight

in lb. per cubic foot of steam of pressure P B, the weight of steam accounted for by the indicator will be $13{,}750\,\frac{r\,W}{P}$ lb., P being the M.E.P.

For diagrams from engines with a late cut-off, or which show only a small amount of compression, the foregoing method is inapplicable, and the following may be substituted. Express the length P Q as a percentage of the whole stroke L, add the percentage of clearance volume Q C, and multiply by the weight W of a cubic foot of steam of pressure P B. From this subtract the product found by multiplying the percentage of clearance by the weight w of a cubic foot of steam at the pressure at the end of the compression as at k. Multiplying this result by 137·5, and dividing by the M.E.P., will give the weight in pounds of steam per I.H.P. per hour accounted for by the diagram.

For multi-cylinder engines, the M.E.P. employed in these calculations is to be that of all the cylinders referred to the particular cylinder from which the card under consideration has been taken.

APPENDIX.

TABLE III.—MEAN PRESSURE FACTORS.

Ratio of Expansion.	Isothermal.	Saturation.	Adiabatic.	Ratio of Expansion.	Isothermal.	Saturation.	Adiabatic.	Ratio of Expansion.	Isothermal.	Saturation.	Adiabatic.	Ratio of Expansion.	Isothermal.	Saturation.	Adiabatic.
1·0	1·0	1·0	1·0	5·6	0·486	0·471	0·461	10·2	0·325	0·310	0·299	14·8	0·249	0·234	0·223
1·2	0·985	0·985	0·984	5·8	0·475	0·460	0·450	10·4	0·321	0·306	0·295	15·0	0·247	0·232	0·221
1·4	0·955	0·952	0·950	6·0	0·465	0·450	0·438	10·6	0·317	0·302	0·291	15·2	0·245	0·230	0·219
1·6	0·919	0·915	0·911	6·2	0·455	0·440	0·429	10·8	0·313	0·298	0·287	15·4	0·242	0·227	0·217
1·8	0·882	0·875	0·871	6·4	0·446	0·430	0·419	11·0	0·309	0·294	0·282	15·6	0·240	0·225	0·215
2·0	0·846	0·840	0·833	6·6	0·437	0·421	0·409	11·2	0·305	0·290	0·279	15·8	0·238	0·223	0·213
2·2	0·813	0·805	0·798	6·8	0·429	0·413	0·401	11·4	0·301	0·286	0·275	16·0	0·236	0·221	0·211
2·4	0·781	0·771	0·764	7·0	0·421	0·405	0·393	11·6	0·298	0·283	0·272	16·5	0·231	0·216	0·206
2·6	0·752	0·740	0·733	7·2	0·413	0·397	0·385	11·8	0·294	0·279	0·268	17·0	0·226	0·211	0·201
2·8	0·725	0·713	0·705	7·4	0·406	0·390	0·377	12·0	0·290	0·275	0·264	17·5	0·221	0·206	0·197
3·0	0·699	0·687	0·678	7·6	0·399	0·383	0·370	12·2	0·287	0·272	0·261	18·0	0·216	0·202	0·192
3·2	0·676	0·664	0·654	7·8	0·392	0·376	0·363	12·4	0·283	0·268	0·257	19·0	0·208	0·193	0·183
3·4	0·654	0·642	0·632	8·0	0·385	0·369	0·357	12·6	0·280	0·265	0·254	20·0	0·200	0·186	0·177
3·6	0·633	0·621	0·612	8·2	0·379	0·364	0·350	12·8	0·277	0·262	0·251	21·0	0·192	0·178	0·170
3·8	0·614	0·601	0·591	8·4	0·373	0·358	0·344	13·0	0·274	0·259	0·248	22·0	0·186	0·172	0·164
4·0	0·596	0·582	0·571	8·6	0·368	0·352	0·338	13·2	0·271	0·256	0·245	23·0	0·180	0·167	0·158
4·2	0·579	0·566	0·553	8·8	0·361	0·346	0·332	13·4	0·268	0·253	0·242	24·0	0·174	0·161	0·153
4·4	0·564	0·550	0·537	9·0	0·355	0·339	0·327	13·6	0·265	0·250	0·239	25·0	0·169	0·156	0·148
4·6	0·549	0·535	0·522	9·2	0·351	0·335	0·322	13·8	0·262	0·247	0·236	26·0	0·164	0·152	0·144
4·8	0·535	0·521	0·508	9·4	0·345	0·329	0·317	14·0	0·260	0·245	0·234	27·0	0·160	0·148	0·140
5·0	0·522	0·506	0·495	9·6	0·340	0·324	0·312	14·2	0·257	0·242	0·231	28·0	0·155	0·143	0·136
5·2	0·509	0·499	0·484	9·8	0·335	0·319	0·307	14·4	0·254	0·239	0·228	29·0	0·151	0·139	0·132
5·4	0·497	0·482	0·472	10·0	0·330	0·314	0·303	14·6	0·251	0·236	0·225	30·0	0·147	0·135	0·127

To find the Mean Pressure :—Multiply the initial pressure by the factor corresponding to the ratio of expansion and to the condition of expansion (see pp. 11-14).

TABLE IV.—PISTON CONSTANTS.

Cylinder Diam. in Inches.	Piston Speed—Feet Per Minute.								
	100	200	300	400	500	600	700	800	900
6	0·086	0·172	0·257	0·343	0·428	0·514	0·600	0·686	0·871
7	0·116	0·233	0·350	0 466	0·583	0·699	0·816	0·932	1·049
8	0·152	0·305	0·457	0·609	0·761	0·914	1·066	1·218	1·370
9	0·193	0·385	0·578	0·771	0·964	1·157	1·348	1·542	1·735
10	0·238	0·476	0·714	0·952	1·190	1·428	1·666	1·904	2·142
11	0·288	0·576	0·864	1·152	1·440	1·728	2·016	2·304	2·592
12	0·343	0·685	1·028	1·371	1·713	2·056	2·399	2·742	3·084
13	0·402	0·804	1·207	1·609	2·011	2·413	2·815	3·218	3·620
14	0·466	0·933	1·399	1·866	2·332	2·799	3·265	3·732	4·198
15	0·535	1·071	1·606	2·142	2·677	3·213	3·748	4·284	4·819
16	0·609	1·219	1·828	2·437	3·046	3·656	4·265	4·874	5·484
17	0·688	1·376	2·063	2·751	3·439	4·127	4·815	5·502	6·190
18	0·771	1·542	2·313	3·084	3·855	4·627	5·398	6·169	6·940
19	0·859	1·718	2·578	3·437	4·296	5·155	6·014	6·874	7·733
20	0·952	1·904	2·856	3·808	4·760	5·712	6·664	7·616	8·568
21	1·050	2·099	3·149	4·198	5·248	6·298	7·347	8·398	9·446
22	1·152	2·304	3·456	4·607	5·759	6·911	8·063	9·215	10·367
23	1·259	2·518	3·777	5·036	6·295	7·554	8·813	10 027	11·331
24	1·371	2·742	4·112	5·484	6·854	8·225	9·590	10 967	12·338
25	1·487	2·975	4·462	5·950	7·437	8·925	10·412	11·900	13·387
26	1·609	3·218	4·827	6 436	8·044	9·653	11·262	12 871	14·480
27	1·735	3·470	5·205	6 940	8·675	10·410	12·145	13·880	15·615
28	1·866	3·732	5·598	7·464	9·329	11·195	13·061	14·927	16·793
29	2·002	4·003	6·005	8·006	10·008	12·010	14·011	16·013	18·014
30	2·142	4·284	6·426	8·568	10·710	12 852	14·994	17·136	19·278
31	2·287	4·574	6·862	9·149	11·436	13·723	16·010	18·298	20·585
32	2·437	4·874	7·311	9·748	12·185	14·628	17·060	19·497	21·934
33	2·592	5·184	7·775	10·367	12·959	15·551	18·143	20·734	23·326
34	2·751	5·503	8·254	11·005	13·756	16·508	19·259	22·010	24·762
35	2·915	5·831	8·764	11·662	14·577	17·493	20·408	23·324	26·239
36	3·084	6·169	9·253	12·338	15·422	18·507	21·591	24·676	27·760
37	3·258	6·516	9·775	13·033	16·291	19·549	22·807	26·066	29·324
38	3·437	6·873	10·310	13·747	17·183	20·620	24·057	27·494	30·930
39	3·620	7·240	10·860	14·480	18·100	21·720	25·340	28·960	32·580
40	3·808	7·616	11·424	15·232	19·040	22·848	26·656	30·464	34·272
41	4·001	8·002	12·003	16·003	20·004	24·005	28·006	32·007	36·007
42	4·198	8·397	12·595	16·793	20·991	25·190	29·388	33·586	37·785
43	4·401	8·801	13·202	17 602	22·003	26·404	30·804	35·205	39·605
44	4·608	9·215	13·823	18·431	23·038	27·646	32·254	36·862	41·469
45	4·819	9·639	14·458	19·278	24·097	28·917	33·736	38·556	43·375
46	5·036	10·072	15·108	20·144	25·180	30·217	35·253	40·289	45·325
47	5·257	10·515	15·772	21·030	26·287	31·544	36·802	42·059	47·317
48	5·483	10·967	16·450	21·934	27·417	32·901	38·384	43·868	49·351
49	5·714	11·429	17·143	22·858	28·572	34·286	40·001	45·715	51·430
50	5·950	11·900	17·850	23·800	29·750	35·700	41·650	47·600	53·550
51	6·190	12·381	18·571	24·762	30·952	37·142	43·333	49·523	55·714
52	6·435	12·871	19·306	25·742	32·177	38·613	45·048	51·484	57·919
53	6·685	13·371	20·056	26·742	33·427	40·112	46·798	53·483	60·169
54	6·940	13·880	20·820	27·760	34·700	41·641	48·581	55·521	62·461
55	7·199	14·399	21·598	28·798	35·997	43·197	50·396	57·596	64·795
56	7·464	14·927	22·391	29·855	37·318	44·782	52·246	59·710	67·173
57	7·733	15·465	23·108	30·930	38·663	46·396	54·128	61·861	69·593
58	8·006	16·013	24·019	32 025	40·031	48·038	56·044	64·050	72·057
59	8·285	16·570	24·854	33·139	41·424	49·709	57·994	66·278	74·563
60	8·568	17·136	25·704	34·272	42·840	51·408	59·976	68·544	77·112

TABLE V.—AREAS OF CIRCLES.

Dia. In.	Area. Sq. In.	Dia. In.	Area. Sq. In.	Dia. In.	Area. Sq. In.	Dia. In.	Area. Sq. In.	Dia. In.	Area. Sq. In.	Dia. In.	Area. Sq. In.
⅛	0·0122	10	78·540	20	314·16	30	706·86	50	1963·5	70	3848·5
¼	0·0490	¼	82·516	¼	322·06	½	730·62	½	2003·0	71	3959·2
½	0·1963	½	86·590	½	330·06	31	754·77	51	2042·8	72	4071·5
¾	0·4418	¾	90·763	¾	338·16	½	779·31	½	2083·1	73	4185·4
1	0·7854	11	95·033	21	346·36	32	804·25	52	2123·7	74	4300·8
¼	1·2272	¼	99·402	¼	354·66	½	829·58	½	2164·8	75	4417·9
½	1·7671	½	103·87	½	363·05	33	855·30	53	2206·2	76	4536·5
¾	2·4053	¾	108·43	¾	371·54	½	881·41	½	2248·0	77	4656·6
2	3·1416	12	113·10	22	380·13	34	907·92	54	2290·2	78	4778·4
¼	3·9761	¼	117·86	¼	388·82	½	934·82	½	2332·8	79	4901·7
½	4·9087	½	122·72	½	397·61	35	962·11	55	2375·8	80	5026·5
¾	5·9396	¾	127·68	¾	406·49	½	989·80	½	2419·2	81	5153·0
3	7·0686	13	132·73	23	415·48	36	1017·9	56	2463·0	82	5281·0
¼	8·2958	¼	137·89	¼	424·56	½	1046·3	½	2507·2	83	5410·6
½	9·6211	½	143·14	½	433·74	37	1075·2	57	2551·8	84	5541·8
¾	11·045	¾	148·49	¾	443·01	½	1104·5	½	2596·7	85	5674·5
4	12·566	14	153·94	24	452·39	38	1134·1	58	2642·1	86	5808·8
¼	14·186	¼	159·48	¼	461·86	½	1164·2	½	2687·8	87	5944·7
½	15·904	½	165·13	½	471·44	39	1194·6	59	2734·0	88	6082·1
¾	17·721	¾	170·87	¾	481·11	½	1225·4	½	2780·5	89	6221·1
5	19·635	15	176·71	25	490·87	40	1256·6	60	2827·4	90	6361·7
¼	21·648	¼	182·65	¼	500·74	½	1288·2	½	2874·8	91	6503·9
½	23·758	½	183·69	½	510·71	41	1320·3	61	2922·5	92	6647·6
¾	25·967	¾	194·83	¾	520·77	½	1352·7	½	2970·6	93	6792·9
6	28·274	16	201·06	26	530·93	42	1385·4	62	3019·1	94	6939·8
¼	30·680	¼	207·39	¼	541·19	½	1418·6	½	3068·0	95	7088·2
½	33·183	½	213·82	½	551·55	43	1452·2	63	3117·2	96	7238·2
¾	35·785	¾	220·35	¾	562·00	½	1486·2	½	3166·9	97	7389·8
7	38·485	17	226·98	27	572·56	44	1520·5	64	3217·0	98	7543·0
¼	41·282	¼	233·71	¼	583·21	½	1555·3	½	3267·5	99	7697·7
½	44·179	½	240·53	½	593·96	45	1590·4	65	3318·3	100	7854·0
¾	47·173	¾	247·45	¾	604·81	½	1626·0	½	3369·6	101	8011·8
8	50·265	18	254·47	28	615·75	46	1661·9	66	3421·2	102	8171·3
¼	53·456	¼	261·59	¼	626·80	½	1698·2	½	3473·2	103	8332·3
½	56·745	½	268·80	½	637·94	47	1734·9	67	3525·7	104	8494·9
¾	60·132	¾	276·12	¾	649·18	½	1772·1	½	3578·5	105	8659·0
9	63·617	19	283·53	29	660·52	48	1809·6	68	3631·7	106	8824·7
¼	67·201	¼	291·04	¼	671·96	½	1847·5	½	3685·3	107	8992·0
½	70·882	½	298·65	½	683·49	49	1885·7	69	3739·3	108	9160·8
¾	74·662	¾	306·35	¾	695·13	½	1924·4	½	3793·7	110	9503·3

TABLE VI.—PROPERTIES OF SATURATED STEAM.

Absolute Pressure. Lb. per Sq. In.	Temperature. Degrees F.	Total Heat. Thermal Units per lb. from 32° F.	Latent Heat. Thermal Units per lb.	Cubic ft. per lb.	Weight of 1 Cubic ft. in lb.	Cubic ft. of Steam from 1 Cubic ft. of Water.
1	102·1	1112·5	1042·9	330·36	0·0030	20,600
2	126·3	1119·7	1025·8	172·08	0·0058	10,730
3	141·6	1124·6	1015·0	117·52	0·0085	7,327
4	153·1	1128·1	1005·8	89·62	0·0112	5,589
5	162·3	1130·9	1000·3	72·66	0·0138	4,530
6	170·2	1133·3	994·7	61·21	0·0163	3,816
7	176·9	1135·3	990·0	52·94	0·0189	3,301
8	182·9	1137·2	985·7	46·69	0·0214	2,911
9	188·3	1138·8	981·9	41·79	0·0239	2,626
10	193·3	1140·3	978·4	37·84	0·0264	2,360
11	197·8	1141·7	975·2	34·62	0·0289	2,157
12	202·0	1143·0	972·2	31·88	0·0314	1,988
13	205·9	1144·2	969·4	29·57	0·0338	1.844
14	209·6	1145·3	966·8	27·61	0·0362	1,721
15	213·1	1146·4	964·3	25·85	0·0387	1,611
16	216·3	1147·4	962·1	24·32	0·0411	1,516
17	219·6	1148·3	959·8	22·96	0·0435	1,432
18	222·4	1149·2	957·7	21·78	0·0459	1,357
19	225·3	1150·1	955·7	20·70	0·0483	1,290
20	228·0	1150·9	952·8	19·72	0·0507	1,229
21	230·6	1151·7	951·3	18·84	0·0531	1,174
22	233·1	1152·5	949·9	18·03	0·0555	1,123
23	235·5	1153·2	948·5	17·26	0·0580	1,075
24	237·8	1153·9	946·9	16·64	0·0601	1,036
25	240·1	1154·6	945·3	15·99	0·0625	[illegible]
26	242·3	1155·3	943·7	15·38	0·0650	958
27	244·4	1155·8	942·2	14·86	0·0673	926
28	246·4	1156·4	940·8	14·37	0·0696	895
29	248·4	1157·1	939·4	13·90	0·0719	866
30	250·4	1157·8	937·9	13·46	0·0743	838
31	252·2	1158·4	936·7	13·05	0·0766	813
32	254·1	1158·9	935·3	12·67	0·0789	789
33	255·9	1159·5	934·0	12·31	0·0812	767
34	257·6	1160·0	932·8	11·97	0·0835	746
35	259·3	1160·5	931·6	11·65	0·0858	726
36	260·9	1161·0	930·5	11·34	0·0881	707
37	262·6	1161·5	929·3	11·04	0·0905	688
38	264·2	1162·0	928·2	10·76	0·0929	671
39	265·8	1162·5	927·1	10·51	0·0952	655
40	267·3	1162·9	926·0	10·27	0·0974	640
41	268·7	1163·4	924·9	10·03	0·0996	625
42	270·2	1163·8	923·9	9·81	0·1020	611
43	271·6	1164·2	922·9	9·59	0·1042	598
44	273·0	1164·6	921·9	9·39	0·1065	585
45	274·4	1165·1	920·9	9·18	0·1089	572
46	275·8	1165·5	919·9	9·00	0·1111	561
47	277·1	1165·9	919·0	8·82	0·1133	550
48	278·4	1166·3	918·1	8·65	0·1156	539
49	279·7	1166·7	917·2	8·48	0·1179	529

TABLE VI.—PROPERTIES OF SATURATED STEAM.

Continued.

Absolute Pressure. Lb. per Sq. In.	Temperature. Degrees F.	Total Heat. Thermal Units per lb. from 32° F.	Latent Heat. Thermal Units per lb.	Cubic ft. per lb.	Weight of 1 Cubic ft. in lb.	Cubic ft. of Steam from 1 Cubic ft. of Water.
50	281·0	1167·1	916·3	8·31	0·1202	518
52	283·5	1167·9	914·5	8·04	0·1246	500
54	285·9	1168·6	912·8	7·74	0·1291	482
56	288·2	1169·3	911·2	7·48	0·1336	466
58	290·4	1170·0	909·6	7·24	0·1380	451
60	292·7	1170·7	908·0	7·01	0·1425	437
62	294·8	1171·4	906·4	6·81	0·1469	424
64	296·9	1172·0	904·9	6·60	0·1516	411
66	299·0	1172·6	903·5	6·41	0·1560	399
68	300·9	1173·2	902·1	6·23	0·1605	388
70	302·9	1173·8	900·8	6·07	0·1648	378
72	304·8	1174·3	899·6	5·91	0·1692	368
74	305·6	1174·9	898·2	5·76	0·1736	358
76	308·4	1175·4	896·8	5·61	0·1782	349
78	310·2	1176·0	895·5	5·48	0·1826	341
80	312·0	1176·5	894·3	5·35	0·1869	333
82	313·6	1177·1	893·1	5·23	0·1913	325
84	315·3	1177·6	892·0	5·11	0·1957	318
86	316·9	1178·1	890·8	5·00	0·2002	311
88	318·6	1178·6	889·6	4·89	0·2044	305
90	320·2	1179·1	888·5	4·79	0·2089	298
92	321·7	1179·5	887·3	4·69	0·2133	292
94	323·3	1180·0	886·3	4·60	0·2176	286
96	324·8	1180·5	885·2	4·51	0·2219	281
98	326·3	1181·0	884·1	4·42	0·2263	275
100	327·9	1181·4	883·1	4·33	0·2307	270
105	331·3	1182·4	880·7	4·14	0·2414	257
110	334·6	1183·5	878·3	3·97	0·2521	247
115	338·0	1184·5	875·9	3·80	0·2628	237
120	341·1	1185·4	873·7	3·65	0·2738	227
125	344·2	1186·4	871·5	3·51	0·2845	219
130	347·2	1187·3	869·4	3·38	0·2955	211
135	350·1	1188·2	867·4	3·27	0·3060	203
140	352·9	1189·0	865·4	3·16	0·3162	197
145	355·6	1189·9	863·5	3·06	0·3273	190
150	358·3	1190·7	861·5	2·96	0·3377	184
155	361·0	1191·5	859·7	2·87	0·3484	179
160	363·4	1192·3	857·9	2·79	0·3590	174
165	366·0	1192·9	856·2	2·71	0·3695	169
170	368·2	1193·7	854·5	2·63	0·3798	164
175	370·8	1194·4	852·9	2·56	0·3899	159
180	372·9	1195·1	851·3	2·49	0·4009	155
190	377·5	1196·5	848·0	2·37	0·4222	148
200	381·7	1197·8	845·0	2·26	0·4431	141
210	386·0	1199·1	841·9	2·16	0·4634	135
220	389·9	1200·3	839·2	2·06	0·4842	129
230	393·8	1201·5	836·4	1·98	0·5052	123
240	397·5	1202·6	833·8	1·90	0·5248	119
250	401·1	1203·7	831·2	1·83	0·5464	114

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